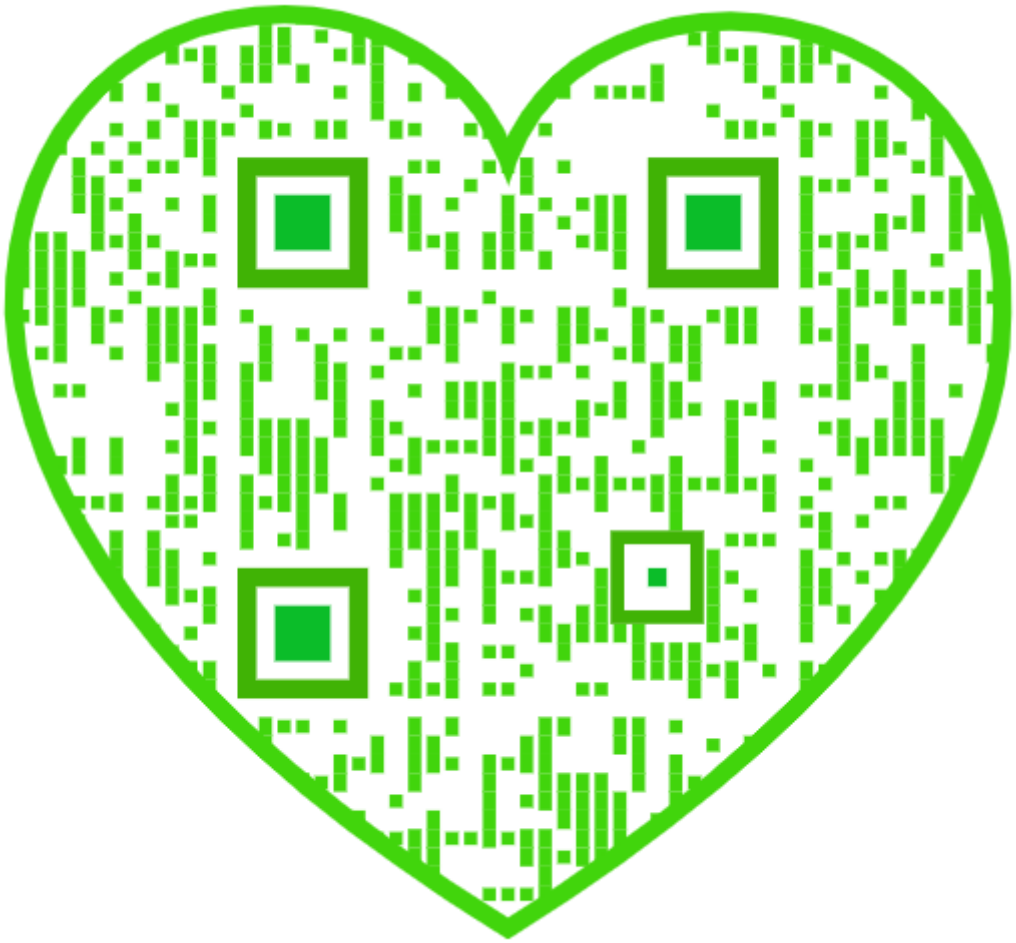


STEAM

AND THE

MARINE STEAM-ENGINE

BY

JOHN YEO

FLEET ENGINEER, ROYAL NAVY; FELLOW OF THE ROYAL SCHOOL OF NAVAL ARCHITECTURE AND MARINE ENGINEERING; MEMBER OF THE INSTITUTION OF NAVAL ARCHITECTS; INSTRUCTOR IN STEAM AND MARINE ENGINEERING, ROYAL NAVAL COLLEGE

London

MACMILLAN AND CO.

AND NEW YORK

1894

All rights reserved

35003

TJF
·Y4

6966194

PREFACE

THIS book has been prepared from my lecture notes, and represents, in an abbreviated form, a considerable part of the course of instruction for Executive Officers at the Royal Naval College. It is intended for Naval Officers, and for Students of Engineering in the earlier part of their training; it is hoped it may also prove useful to Officers of the Mercantile Marine and others desirous of obtaining some knowledge of the propelling machinery of ships, and of various matters connected with its use and management. The special aim of the book is to give a sound general knowledge of the subject, with as much detail as appears to be necessary for the end in view, in as brief and simple a manner as possible. The subject is a large one, and in order to keep the matter within a small compass it has necessarily been much condensed; this is especially the case in the parts dealing with principles, which I would have much preferred to write with greater fulness. What is given, however, is the result of careful consideration; and I hope the brevity of the statements made may not prevent them from being sufficiently clear to be easily understood.

The illustrations are, in part, reproductions of lecture diagrams, but are principally from drawings specially made for the book. The design has been to secure clearness of illustration in regard to the principal features rather than completeness of detail, and

in some cases simple elementary sketches have been preferred to, or have been given with, the more detailed drawings. The illustration of the Yarrow water-tube boiler, Fig. 17, has been reproduced, on a reduced scale, from *Engineering*, vol. li. p. 79; I am indebted to the courtesy of the Editor of that journal for permission to insert it. I am similarly indebted to the firm of Messrs. Maudslay, Sons, and Field for permission to reproduce the drawing of the Belleville boiler, given on p. 31, from their pamphlet on the subject; and to Messrs. Thornycroft and Co. for a photograph from which the drawing of the Thornycroft boiler, reproduced on p. 28, was made.

I have availed myself of the information published in papers read before the Institution of Naval Architects and other societies, especially those read by professional Officers of the Admiralty.

As one of Professor Cotterill's old pupils, I take this opportunity of acknowledging my indebtedness to him.

I wish, further, to acknowledge my obligation to my colleague, Mr. C. W. J. Bearblock, R.N., for much kind assistance, especially in regard to the drawings.

ROYAL NAVAL COLLEGE,
October 1894.

CONTENTS

CHAPTER I

INTRODUCTION

PAGE

Steam-Engine—General Features 1
Application of Steam Power to Propulsion—Early Progress . . . 2
Forms of Engines—Paddle, Screw 3
Advance in Marine Engineering. Economy of Fuel—Economy of Weight and Space—Changes leading to improvement in these particulars . . 5

CHAPTER II

BOILERS

Requirements—Conditions of Working—Working Pressure and Test Pressure—Material 9
Tubular Boiler. General Description—Action 10
Heating Surface 11
Furnace Arrangements 12
Form of Boiler Shell in relation to Pressure—Strength of Circular Section . 13
High Cylindrical Boiler. Features—Construction and Strength—End-Plates—Furnaces, Plain and Corrugated—Combustion-Chambers—Tubes . 15
Staying—Parts Stayed and Description of Stay used 19
Oval Boiler—Double-ended Boiler 21
Low Cylindrical Boiler. Features and Construction 23
Particulars of Ordinary Naval Boilers 25
Locomotive Boiler. Special Requirements—Features and Construction . 25
Water-Tube Boilers. General Features and Advantages of Class—Thornycroft, Yarrow, and Belleville Boiler 27
Forced Draught applied to Ordinary Boilers. Object—System employed in Service—Arrangements—Air-Pressure—Results—Admiralty Ferrule . 32
Boiler Mountings. Safety-Valves—Stop-Valves—Pressure-Gauges—Feed-Valves—Water-Gauges and Test-Cocks—Blow-out Valves . . 37
Main Steam-pipe Connections—Arrangement and Object 45
Care of Boilers under Weigh 47

CHAPTER III

ENGINES

PAGE

Rotatory Motion. Action of Crank and Connecting-Rod—Irregularity of Turning, Effects of and how reduced—Necessity for Guide—Influence of relative length in Connecting-Rod 50
Arrangements of Working Parts. Horizontal Screw Engines: Direct-acting Type—Return-Connecting-Rod Type—Vertical Screw Engines: Type and General Features 53
Twin-Screw Engines—General Arrangement in War-ships 59
Paddle Engines 59

CHAPTER IV

DETAILS OF WORKING PARTS

Cylinder Construction. Shell—Cover—Face—Liner—Fitting of Liner, Advantages 60
Cylinder Fittings. Escape-Valves—Drain-Valves—Jacket Stop-Valve—Indicator Pipe—Starting-Valve 63
Piston. Requirements—Construction—Body—Packing-Ring and Springs—Junk-Ring—Guard-Ring—Ramsbottom Rings 64
Piston-Rod, Securing of 68
Stuffing-Box, Ordinary Construction 68
Packing—Metallic Packing for Cylinder Boxes 69
Adjusting Gear for Glands 70
Framing for Vertical Engines 71
Piston-Rod Head—Connecting-Rod—Gudgeon Bearing—Crank-end Bearing—Bearing Brasses—White Metal 72
Piston-Rod Guide—Guide-Plate—Slipper 74
Shafting and Bearings, Arrangement of, in Twin-Screw Ships—Hollow Shafting—Details of Crank 75
Shaft Coupling—Stress in Shaft greater aft than forward—Main Bearing . 77
Bearings. Conditions for good working—Lubrication—Centrifugal Lubricator 78
Screw-Shaft Bearings—Stern-Tube—Lignum Vitæ Bearings in Stern-Tube and Outer Bracket, with Water Lubrication 80
Thrust-Bearing. Purpose—Simple form—Adjustment—Usual form, with independent adjustment for each collar 83

CHAPTER V

SLIDE-VALVE

Office—Necessary Changes in Communications—Action of simple Slide-Valve without Lap or Lead, and Defects—Requirements for Good Working . 86
Action of Valve with Lap and Lead—Definitions: Lap, Lead, Angular Advance of Eccentric—Summary—Statement of Slide-Valve Operations, how produced, and effect 88

PAGE

Obliquity of Connecting-Rod, Effect on Steam Distribution and on Valve Setting in Vertical Engines 89

Improvements for ease in working of Slide-Valves—Double or Treble-ported Flat-faced Valve—Equilibrium or Relief Ring, Descriptions of—Cylindrical or Piston Slide-Valve—Effects, to shorten stroke and reduce travel—Balance Cylinder and Piston 92

CHAPTER VI

Slide-Valve Gear—Starting and Reversing Arrangements

Eccentric and Rod. Purpose of Mechanism—Action—Modification of Crank and Rod—Detail 98

Link-Motion for Reversal—Positions of Eccentric suitable for Ahead and Astern running—Double Eccentrics and Link, Action in Full Gear—Linking-up and Centring the Links, Effects of 99

Forms of Link and Details—Curvature of Link 103

Reversing Gear. Description ordinarily used—Defects of old arrangement—"All-Round" arrangement of Gear, Advantage—Other Reversing Arrangements 105

Special Expansive Gear—Expansion Valve—Gear for adjusting the several Links independently, for linked-up working 107

Regulating-Valve and Manœuvring-Valve—Starting-Valves 109

Raising Steam and Warming Engines—Starting, Stopping, and Reversing—Under Weigh at Reduced Speeds, Conditions for Economical Working . 109

CHAPTER VII

Indicator Diagrams

The Indicator, Description of Instrument and Method of using it . . 112

Normal Diagram from Simple Cylinder exhausting to Condenser—Explanation of Form—Names of the several Parts, Conditions indicated by them . 114

Compound Diagrams, Features corresponding with method of using Steam . 116

Measurement of Diagrams for Mean Effective Pressure—Scale of Pressures . 117

CHAPTER VIII

Principles—Properties of Steam

Work, Energy, Mechanical Energy—Measurement of Work and of Energy—One Foot-Pound the Unit of Work or of Energy—Method of estimating Work done in operation of Steam-Engine 119

Power—One Horse-Power the practical Unit of Power—Value of the Horse-Power—Expression for I.H.P. developed in Cylinder of Steam-Engine . 120

Heat, Principal Effects produced by—Latent Heat, Sensible Heat—Measurement of Heat, British Thermal Unit—Mechanical Theory of Heat—Mechanical Equivalent of Heat 121

PAGE

Steam. Processes involved in ordinary operation of Steam-Engine—Generation of Steam—Pressure, Temperature, and Volume per Pound . . 122
Condition of Steam: Saturated, Wet or Dry, and Superheated—Particulars of these Conditions 124
Heat-Expenditure in producing Steam—Slow increase of Total Heat with increase of Pressure 126
Latent Heat of Evaporation, Effects of—External Work . . . 127
Condensation of Steam, how utilised in Steam-Engine, and Practical Effects—Vacuum-Gauge 127
Action of Steam—Expression for Work done in driving Piston—Expansive Property of Steam—Comparison with Action of non-expansive Fluid . 128
Admission Work. Expansive Work—Expansive Work the Source of Economy—Approximate Law of Expansion. 129
Theoretical Indicator Diagram—Benefit of Expansion illustrated . . 131
Efficiency of Steam in Practical Engine—in Theoretically Perfect Engine . 132

CHAPTER IX

Expansive Working in Practice

Single-expansion or Simple Engine—Double-expansion Engine—Triple-expansion Engine—Arrangements providing for Expansion in these Types. 134
Profitable Expansion in any given Cylinder practically limited—Reasons for Limitation—Waste by Action of Internal Surfaces of Cylinder . . 136
Steam-Jacket, Action of—Compound Expansion, Principle of, and Advantages of System 137
Arrangements of Compound-expansion Engines—Double-expansion, Woolf, Two-cylinder, and Three-cylinder Types—Triple-expansion, ordinary Naval Type—Features of these Types 138
Triple-expansion Engines—Cylinder Proportions and Total Expansion less in Navy than in Mercantile Marine—Reasons for Difference, and Effect on Economy 140
Receiver Pressure, Adjustment of—Effect of altering Points of Cut-off in the several Cylinders—Expansion not wholly effective, "Drop"—Division of Power between Cylinders, means of regulating 141
Coal Consumption in Triple-expansion, Double-expansion, and Single-expansion Engines 145
Special Fittings of Compound-expansion Engines. Reducing-Valves for I.P. and L.P. Cylinder Jackets—Safety-Valves for the same Jackets and for Receivers—Pressure-Gauges for Jackets and Receivers . . . 146
Combined Diagrams—Method of Combining and Use of 147

CHAPTER X

Condenser and Fittings

Surface Condenser—Circulating Arrangements — Circulating Pumps and Engines—Sea and Bilge Suctions 152
Condenser Detail 156

PAGE
Air-Pumps—Office, Construction, and Action 157
Feed-Tank and Fittings—Reserve Feed-Tanks—Feed-Pumps, Suctions, and Deliveries 160
Course of Working Fluid as Feed-Water from Condenser to Boilers, and as Steam through Engines to Condenser 162

CHAPTER XI

Feed-Water for Boilers—Boiler Preservation

Reasons for avoiding use of Sea Water in High-pressure Boilers—Salts in Sea Water—Sulphate of Lime the principal Scale-forming Ingredient—Solubility dependent on Temperature—Insoluble at High Temperatures—Low-pressure Boilers, Method of working—High-pressure Boilers, Fresh Water necessary 162
Arrangements for minimising Waste of Fresh Water, and making up Unavoidable Waste—Auxiliary Condensers—Silent Blow-off—Feed-Tanks as Reservoirs, with Overflow to Reserve Tanks—Preservation of Drainage Water 165
Make-up required in Ordinary Cases—Distilling Arrangements—Double Distillation—Weir Evaporator and Normandy Condenser . . 166
Boiler Preservation. Brief Statement of Main Facts—Causes of Deterioration in Boilers In Use, and in Boilers Out of Use, and Precautions Observed . 169

CHAPTER XII

Combustion and Fuel—Efficiency

Combustible Constituents of Fuel—Combining Proportions and Calorific Value of Hydrogen and Carbon—Incomplete and Complete Combustion of Carbon 171
Constituents of Coal—Combustion of Coal in Furnaces: 1. Hydro-Carbons, Smoke Production, Air Admission; 2. Uncombined Carbon . . 172
Varieties of Coal—British Steam Coals, Proportions of Hydro-Carbons, Evaporative Value and other Qualities 174
Patent Fuel—Liquid Fuel 176
Heat-Value of Coal—Proportion of Energy utilised in developing Power in Marine Engines—Proportion of Power utilised in Propulsion—Efficiency of Boiler, of Steam, of Mechanism, and of Screw—Distribution and Principal Causes of Loss 177

CHAPTER XIII

Screw-Propeller and Propulsion

Introduction of Screw-Propeller—Length, Number of Blades, Diameter, and Pitch of Screw—Griffith's Screw, Features—Usual Form and Construction of Screw—Variation of Pitch 180
Action of Screw-Propeller—Slip: Apparent, Real 184

PAGE

Twin-Screws, Advantages—Screw and Paddle-Wheel—Augment of Resistance 186

Resistance to Ship's Progress—Frictional Resistance, Variation with Speed—Wave-making Resistance 188

Relations, at Moderate Speeds, between Power, Speed, and Consumption: (*a*) Power and Speed, (*b*) Consumption per Hour and Speed, (*c*) Consumption per Knot and Speed, (*d*) Consumption, Speed, and Distance—Curve of I.H.P. and Speed—Coal for Auxiliary Purposes 188

Most Economical Speed—Determination by Trial—Graphic Method . . 193

STEAM AND THE MARINE STEAM-ENGINE

CHAPTER I

INTRODUCTION

THE steam-engine of the present day, in its ordinary form, has been produced by the gradual development and improvement of Watt's reciprocating, double-acting, condensing, and rotative engine. Though greatly improved in all respects since the time of Watt, the engine remains in its essential features practically unaltered. These features, very briefly described, are as follows:—

Steam is used to drive a piston forward and backward along the bore of a cylinder, through a definite stroke which is determined by the connection of the piston with the mechanism outside the cylinder. The steam is admitted alternately at the two ends of the cylinder for successive strokes, and the opposite end to that at which the steam is being admitted for any given stroke is opened freely to exhaust.

Each stroke is thus produced in the same manner, viz. by the exertion of pressure on one face of the piston while exhaust takes place from the opposite face; and the steam and exhaust communications are interchanged, accordingly, for each fresh stroke of the piston.

Reciprocating and Double-Acting.—The steam-engine is thus a reciprocating engine so far as the original production of motion is concerned; it is also double-acting, which term distinguishes it from Watt's earlier engine, known as the single-acting engine, in which the piston was driven by the steam in one

direction only, and was returned to its starting-point by other means. The single-acting engine was especially adapted for certain kinds of work in which an intermittent exertion of power sufficed.

Condensing.—The steam, when its work is done, is allowed to escape from the cylinder, but not usually to the atmosphere; it passes generally to an exhausted condensing vessel, where it is immediately condensed to water, and the resistance the exhaust steam would otherwise offer to the piston's motion is thus largely removed. In certain cases, as in locomotive engines, the condenser is omitted for special reasons. The great majority of stationary land engines, however, and all marine engines, are condensing engines.

Rotative.—The most convenient motion for general purposes, and the one most frequently required in machinery, is that of continuous rotation, from which indeed other kinds of motion may easily be obtained; and this the steam-engine is adapted to produce. The reciprocating motion of the piston produced by the action of the steam is converted into the rotatory motion of a line of shafting by the simple mechanism of the crank and connecting-rod, provision being made for overcoming the difficulty arising from the *dead points* of the crank. The engine being thus double-acting and rotative, there is *practically a sustained effort resulting in continuous rotation.* This renders the steam-engine directly applicable to the general purposes for which power is required; and, in particular, to the purposes of land locomotion and marine propulsion.

For very many purposes a single engine may be used, the difficulty of the dead points being, in most cases, overcome by a fly-wheel carried on the crank shaft. In locomotive and marine work, however, where readiness of starting and reversing are of the greatest importance, two engines at least are used to drive the same line of shafting; and the cranks are so placed that the engines help each other over their respective dead points.

Application to Propulsion.—The successful application of steam power to propulsion afloat began practically with the opening years of the present century, about twenty years after the introduction of Watt's double-acting rotative steam-engine. In the original form of that engine the crank and connecting-rod had not been used; and it was only after the substitution of this

mechanism for that originally employed for the same purpose, that the engine became really suitable for propulsion.

Early Progress.—Up to 1820 there were only a few small river steamers at work, engaged in passenger service and towing. More powerful steamers, however, soon appeared, and in rapidly increasing numbers; employed not only on rivers, but on coasting and other voyages of gradually increasing length. The Irish Mail Service and the General Steam Navigation Company were established in the early part of the period 1820-30, and during the latter part of it steam power was introduced into the Navy. Between 1830 and 1840 the use of steam-vessels became very greatly extended. Lines of steamers were established to the Peninsula and the Mediterranean, and finally to India (P. & O. Company) and across the Atlantic (Cunard Line, 1840); while in the Navy a number of steam frigates were built, and the power supplied was gradually increased.

At the close of this period all steam-vessels were still propelled by the paddle-wheel.

Paddle Engines.—The type of engine adopted at first was that known as the side-lever engine, which was merely the marine adaptation of the beam engine then in common use on land, and in which motion was communicated by the piston to one end of a pivoted beam, the other end of which was connected with the crank by the connecting-rod. This engine was widely used for a considerable time even after the introduction of, in some respects, improved types. With many merits, however, this type of engine had the defect of being very heavy, and of occupying great space in comparison with the power developed—a fault which became more serious as the power provided in ships increased. Gradually, therefore, various other types were introduced, all dispensing with the use of beams, and aiming at saving weight and space. Of these, two may be mentioned: the Gorgon Engine and the Oscillating Engine. The first was an example of the simple direct-acting type, in which the piston is directly connected with the crank by a piston-rod and a connecting-rod; the cylinder being, in this particular case, placed vertically below the shaft, as in Fig. 1. This type, placed on its side, has since been largely used as a horizontal screw engine; and, turned upside down, is now in general use for the screw. The second is the best example of a number of types

which, by various means, avoided the difficulty from which the Gorgon Engine suffered, viz. want of space, arising from its vertical position and from the limited height of the paddle-shaft. The simple direct-acting and the oscillating are the only types which have survived for the paddle-wheel; and the former is now

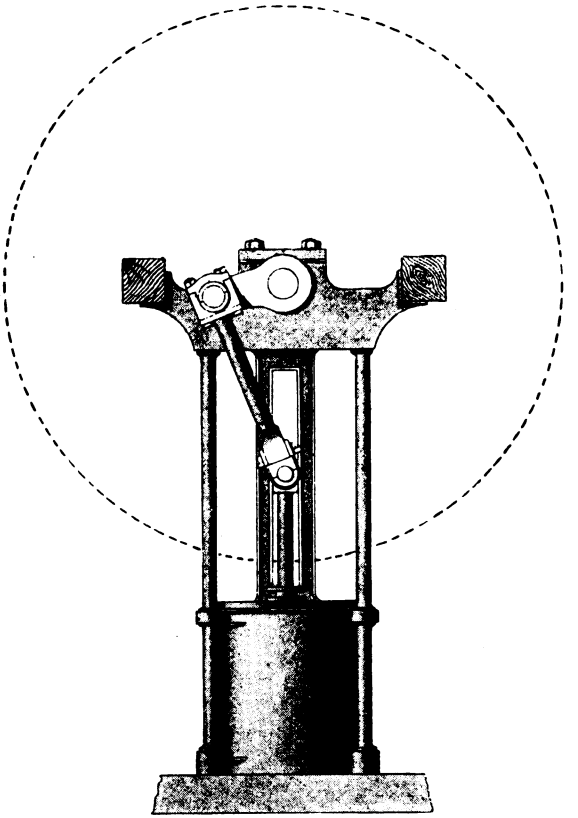


FIG. 1.

placed, not vertically as in the Gorgon Engine, but at a large inclination to the vertical, in order to obtain the space necessary in the direction of the stroke.

Screw Engines.—About 1840 the proposal to use the screw as a propeller attracted great attention, especially after the successful performance of the *Archimedes*, a boat in which the screw was for the first time tested on a moderately large

scale. This was followed by the Admiralty trials of the screw in the *Rattler* in 1844-45, after which the screw propeller came rapidly into use, especially in the Royal Navy, where, apart from its merits as a propeller, it afforded special and conspicuous advantages.

At first the forms of engine employed were those already in use with the paddle-wheel; and, higher revolutions being necessary for the screw, these were produced by means of toothed gearing connecting the engine-shaft with the screw-shaft. Soon, however, an entire change was made in the arrangement of screw engines for men-of-war. Since the screw was submerged and the screw-shaft low down, it became possible by a suitable arrangement of the engines to secure protection for the whole of the machinery by keeping it below the water-line. It was found also that the engine-shaft might be connected directly to the screw-shaft without gearing, and the engines themselves run at the revolutions necessary for the screw without injury.

Horizontal Engines.—This led to the use of fast-running engines placed horizontally. Many forms of horizontal screw engine were tried; but, in a few years, these became reduced to the three well-known types, the Direct-Acting Engine, the Return Connecting-Rod Engine, and the Trunk Engine. With one or other of these three types, for a period of over twenty years, all war-vessels were fitted; and for a further period of about fifteen years, that is up to a few years ago, horizontal engines (of the first two types) continued to be fitted to all unarmoured naval vessels.

Vertical Engines.—About 1871 *Vertical* Direct-Acting Engines were again introduced in connection with twin-screws for armoured vessels, the engines being *inverted*, that is, placed with the cylinders vertically above the shaft, as had long been the practice in the Mercantile Marine. These engines have many advantages, but they cannot well be kept entirely below the water level; their use has therefore, until recently, been confined to armoured vessels.

The fitting of protective decks, however, and, in some cases, of armour protection internally for the upper parts of the engines, as well as the general use of coal protection, has enabled their use to be extended to unarmoured vessels, with the result that twin-screw vertical engines are now fitted to war-ships of all classes.

Advance in Marine Engineering

Apart from improvements in detail, the real advance in marine engineering since the early days has been chiefly in two directions: (1) the engines burn much less fuel in comparison with the power developed than they did formerly; (2) they develop much more power in comparison with their weight and size. In other words, the two principal results secured have been comparative economy of fuel and comparative economy of weight and space.

Economy of Fuel.—The present rate of fuel consumption for a given power is, roughly, one-fourth to one-fifth the consumption in the early engines, one-third the consumption in the early screw engines, and one-half that necessary, say, twenty-five years ago, just before the Compound Engine was adopted in the Navy. This progressive economy has resulted principally from a steady advance to higher steam pressures and increased expansion, in connection with more efficient arrangements for carrying out the expansion. Of these again the increased expansion, and the improved arrangements for expansive working, are the real causes of the economy, the higher steam pressure being merely a necessary condition of the larger expansion, and not in itself a direct source of economy.

In the early marine engines the working pressure was 4 or 5 lbs. only by steam-gauge; the engines used scarcely any expansion, and were fitted with jet condensers, which gave practically a salt-water feed for the boilers. By 1850-55 the working pressure had gradually risen to 20 lbs.; but so long as jet condensers were still used it was found impracticable to carry more than 20 to 25 lbs. on account of the excessive incrustation of the boilers which occurred with higher pressures. The success and general adoption of surface condensers a few years later removed this difficulty; the pressure rose to 30 to 35 lbs., and, on the substitution of the cylindrical form of boiler for the rectangular form, about 1870, to 60 lbs. Since that time there has been a steady advance to the 155 lbs. pressure now usual in Navy boilers; this has been rendered possible, chiefly, by the use of mild steel, by improvements in manufacture, by the use of corrugated furnaces, and by the provision of fresh water for making up waste.

The continual increase of pressure indicated above has been steadily utilised in securing increased expansive working, the most important steps in this connection being the adoption of the Compound Engine, and the comparatively recent change to the Triple-Expansion Engine, which works on the same principle, but is suited for a higher pressure.

Economy of Weight and Space.—In ordinary ships' boilers the reduction of weight and space in comparison with steam-producing power has resulted, principally, from an early change of type from flue to tubular boilers, and, in recent years, from the use of forced draught. The use of steel has lightened the boilers, while the higher pressure carried has had the opposite effect.

In the engines, apart from improvements in detail, weight and space in proportion to power have been reduced, first, by changes of type; second, by using higher pressures and higher piston speeds, the latter produced by higher revolutions. With regard to this point, the power developed in any engine will depend on the product of driving force and piston speed; or, if we take *a cylinder of given diameter and stroke*, the power developed in it will depend on the product of *average effective pressure* and *revolutions*. Both of these factors have been increased; and the power produced with given sized cylinders, pistons, slide-valves, etc., has increased accordingly.

The average pressure has increased with the use of higher boiler pressures, though to a smaller extent than would have been the case if the steam had not been used so expansively. The revolutions have also greatly increased in all classes of vessels.

On the introduction of screw engines acting without gearing, the piston speed, which had been 200 to 250 feet per minute, advanced to 400 feet per minute as the result of increased revolutions; from which point the speed has been gradually increased to 850 or 900 feet, and in special classes to 1000 to 1200 feet per minute, with a corresponding increase in the revolutions.

Thus we find that in paddle engines of the beam type with flue boilers, a ton total weight of machinery produced about 2 H.P.; in the oscillating type with tubular boilers, about 4 H.P.: and in screw engines the figure, beginning with 5, has increased gradually in ordinary types to 8 to 10 with natural draught, and

11 to 13 with forced draught.[1] In special high-speed machinery, such as that used in torpedo-boats, and in a modified form in torpedo gunboats, it varies from about 16 in the latter to nearly 30 in the former, this result being largely due to the use of the locomotive form of boiler worked with forced draught: with water-tube boilers it is higher still, other things remaining the same.

The extreme powers, however, whether in boats or ships, can be obtained only for a short time, and not with economy. The two objects, fuel economy and great power on a given weight, are to some extent inconsistent; the former requires that the boilers shall be working easily, and that considerable expansion shall be used in the engines, both of which conditions interfere with the attainment of extreme power on a given weight.

The gradual reduction of weight of machinery and of coal consumption, in comparison with the power developed, has enabled greater power to be supplied and greater speed obtained in vessels of all classes than would otherwise have been possible; and further, has secured a greatly increased steaming radius at moderate speeds. This applies to cruisers and the smaller classes of vessels even more than to battle-ships; but, taking the latter class for illustration, the power supplied for a given displacement has been about doubled since the date of the early ironclads, and the speed has increased from about 14 to $17\frac{1}{2}$ or 18 knots; yet the weight of propelling machinery carried in a ship of given displacement is not much greater than was necessary for the smaller powers of that date, though it now includes many fittings and much duplication which then did not exist; and a given quantity of coal devoted to propulsion will carry the ship about two-and-a-half times as far at a ten knot speed.

[1] Further particulars will be found in a table published by Mr. A. J. Durston, Engineer-in-Chief, Admiralty, in a paper read before the Institution of Naval Architects (*Transactions*, 1892), entitled "Some Notes on the History, Progress, and Recent Practice in Marine Engineering."

CHAPTER II

BOILERS

Requirements.—The most important requirements of a steam-boiler are, that it should carry steam of high pressure with safety and without leakage; that it should produce a large quantity of steam in a given time; that it should work quietly and steadily, giving steam in a dry condition; that it should give a good evaporative result for the coal burnt, and that it should be accessible internally for examination and cleaning.

Conditions of Working.—The conditions under which boilers work are obviously very trying; they are exposed to great differences of temperature in the several parts, and consequently to strains from unequal expansion, especially during the raising or lowering of steam. They are liable to loss of strength locally from over-heating, which may arise from temporary shortness of water, or from deposit; and to a general or local reduction of strength by corrosion. Further, should the safety-valves fail to act efficiently, they may at any time be subject to over-pressure.

A certain margin of strength in the material, founded on experience, is of course provided in the original construction; but the chief reliance is placed on the hydraulic test which is applied to the boiler as a whole, both before use and periodically afterwards. The boiler is examined for tightness of joints, and is carefully gauged in all accessible parts, while the water pressure is on; and no permanent alteration of form or dimensions should be produced by it. In high-pressure boilers in the Navy, the water-test pressure is 90 lbs. above the *working* or *blowing-off* pressure intended to be carried; by this is meant the pressure to which the safety valves are loaded, which is also the pressure shown by the steam-gauge when the valves are about to lift.

Formerly the rule was to test up to double the working pressure; and this is still the rule for naval boilers working with 90 lbs. pressure and under, and for boilers of the mercantile marine at all pressures. It is obvious, however, that the margin of strength provided by this rule is greater in the case of a high-pressure boiler than in one with lower pressure; and in boilers with more than 90 lbs. working pressure it is considered sufficient in Admiralty practice to test up to the point stated above.

Material.—The material used in modern boiler construction is mild steel; the structure is built up of steel plates riveted together, strengthened where necessary by steel stays; the tubes are also of steel. The requirements in the material are strength and absence of brittleness; wrought iron, which was formerly used, was good in these respects, but mild steel is better; it is stronger than wrought iron by at least 30 per cent, and, if sufficiently mild, is even more ductile.

Tubular Boiler

General Description.—The ordinary marine boiler is of the *tubular* class, and consists of an external *shell* containing the water and the steam, and of internal parts, comprising *furnaces*, *combustion-chambers*, and *tubes*, with various stays and fittings. The parts may be seen in Figs. 3 and 4 and in Fig. 14; in the former the furnaces are marked *f*, the combustion-chambers *c*, the tubes *t*, and the shell S.

The furnaces, within which the combustion of the coal takes place, are situated at the lower part of the boiler, and pass from the front towards the back until they open into the combustion-chambers. These are large structures intended to provide the space necessary for the proper combustion of the furnace gases. From the combustion-chambers tubes pass through the water, either returning to the front or passing on to the back end of the boiler; in either case they open into the *smoke-box*, marked *m* in Fig. 4, which is in communication with the outer air by way of the *uptake*, a casing leading to the funnel.

There is an uptake from each boiler: these are united at the base of the funnel. The funnel causes a rapid flow of the hot gases from furnaces to combustion-chambers, through the tubes to the smoke-box, and so to the outer air; and a natural flow of

air, or draught, to the fires is the result. Each uptake is fitted with a damper by which the particular boiler may be shut off from the funnel when not in use; and, if forced draught is to be used, the uptake is divided into compartments fitted with dampers so that each combustion chamber may be shut off from the funnel while the fire or fires communicating with it are being cleaned—thus preventing the passage of cold air through the boiler at that part, while allowing the draught to the other fires to be maintained. The dampers are closed also while sweeping tubes.

The furnaces, combustion-chambers, and tubes, are surrounded by water, which usually stands at the height of at least 7″ above the combustion-chamber crowns. The metal of these parts, excepting the lower parts of the furnaces and combustion-chambers, exposed to the temperature of the fire or hot gas on one side, and having water in contact with it on the other side, serves to transmit heat to the water, and is therefore called the *heating surface* of the boiler. The tubes divide the gases into a great number of small streams, each stream surrounded by heating surface.

The heat received on the fire side of the heating surface, as the result of radiation or of the contact of flame or of hot gas, and transmitted by conduction to the water lying in contact with the metal, is distributed throughout the body of the water by the process of *convection*; that is to say, a circulation is set up, steam and heated water rising continually from the various parts of the heating surface, and being replaced by cooler water. It is therefore important that the arrangement of the internal parts be such as will not impede this circulation more than is necessary. For this reason the tubes are placed in vertical rows, with not less than 1″ clear between the rows, and are also in nests or batches of tubes, with water spaces between the batches. Other water spaces are provided behind and between the combustion chambers, as shown in the figures.

The various parts of the heating surface are by no means equally active in steam generation; the tubes generate the smallest quantity, and the furnaces the greatest quantity of steam, per square foot of surface in a given time. The furnaces are exposed to the highest temperature, while the hot gases within the tubes have become much reduced in temperature, especially towards the smoke-box end; the transmission of heat is consequently much more rapid through the furnace plates. Further,

the tubes rapidly become more or less fouled by soot, which is a bad conductor of heat. The combustion-chamber plates are intermediate in activity between the tubes and the furnace plates. Notwithstanding their comparative inefficiency, however, the tubes are of great value to the boiler, since they provide frequently as much as $\frac{9}{4}$ of the entire heating surface.

Furnace Arrangements.—The fuel is burnt on a fire-grate, *g*, Figs. 3 and 4, consisting of a number of wrought iron or steel fire-bars lying along the furnace and supported at their ends. The bars are $\frac{3}{4}''$ to $1''$ in width at top; and spaces between them, $\frac{1}{2}''$ to $\frac{5}{8}''$ wide, provide for the passage of air to the fuel. These air spaces are obtained by widening the ends of the bars, or fitting them with stops, *s*, Fig. 2, which act as distance-pieces. The sketches, Fig. 2, show part of the grate in plan and in end view. In naval practice the grate is 6 to 7 feet long, made up of three lengths of fire-bars. It slopes downward from front to back, where it is terminated by the bridge, *r*, Fig. 4, a bank of fire-brick supporting the fuel at the back of the fire. The slope is convenient for firing, and gives more space at the inner end of the furnace for the gases. The outer end of the furnace is closed by a frame, *e*, secured to the boiler front, and carrying the *furnace door*, *d*, for the upper part, or furnace proper, and also the *ash-pit door*, *a*, unless the latter is hung separately. The ash-pit is the space below the grate; and the door at its mouth serves to regulate the supply of air to the fires under ordinary circumstances. In the corrugated furnaces now used, an ash-tray is fitted in the ash-pit to catch the ashes and facilitate their removal: a little water is kept in the tray for quenching them.

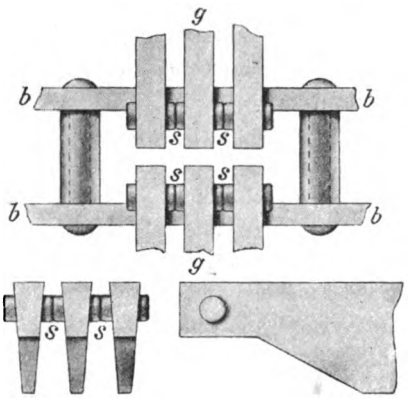


FIG. 2.

The usual construction of the furnace door is shown in Fig. 4. It is in three thicknesses of plate, separated by air spaces; and a passage for air through the door to the furnace proper is provided by means of holes through the central thickness to which the air

has access; in some cases a sliding-shutter arrangement in the outer plate enables the supply to be regulated. This direct air admission is desirable for the combustion of the coal gases; and the construction serves also to keep the door cool and protect it from injury, and to check the radiation of heat into the stokehold.

Immediately within the furnace door is the *dead plate.* On this plate, and on the plate carrying the bridge, the outer and inner ends of the grate are supported; the other ends of the fire-bars rest on *bearing-bars, b,* Figs. 2 and 4, which cross the furnace and are secured to it by brackets.

Form of Boiler Shell.—The boiler shell used for low-pressure boilers was of rectangular section, or box-shaped, more or less modified to suit the form of the vessel. In the modern boiler the form is cylindrical.

The rectangular form was a convenient one, utilising the greater part of the available space for the boilers; but it was essentially weak. With this form the pressure tends to bulge the flat sides, and eventually to drive them apart by destroying their connections: this action would not be prevented by any ordinary increase of thickness, and can only be met by staying the opposite plates together; thus the strength of the structure becomes dependent on the stays. The nearer together these are placed the stronger is the shell, but the less accessible does it become for internal examination and cleaning. This limited the pressure carried in these boilers to 30 to 35 lbs.

A shell of circular section is stronger than the rectangular shell because such a section does not tend to alter form under internal pressure, and the structure will therefore not yield until the tensile strength of the plate is overcome, or the rivets are sheared. With a given material, the strength of this form to resist rupture is in direct proportion to the thickness of plate used. On the other hand, the force tending to produce rupture is in proportion to the product of pressure and diameter. Thus the pressure which can be carried depends directly on the thickness of plate and inversely on the diameter of the section; in other words, it depends on the proportion $\frac{\text{Thickness of Material.}}{\text{Diameter of Shell.}}$

A cylindrical shell, therefore, becomes capable of carrying a higher pressure if either the thickness of plate be increased or the diameter diminished.

Few examples of the rectangular class of boiler still remain, and those only in ships of old types. Many of the more serviceable of the ships so fitted have been, or are being re-engined and supplied with cylindrical boilers; among these are the *Devastation*, *Thunderer*, *Rupert*, and *Monarch*, turret-ships, and the *Hercules*, *Sultan*, and *Superb*, broadside-ships. Cylindrical boilers in the

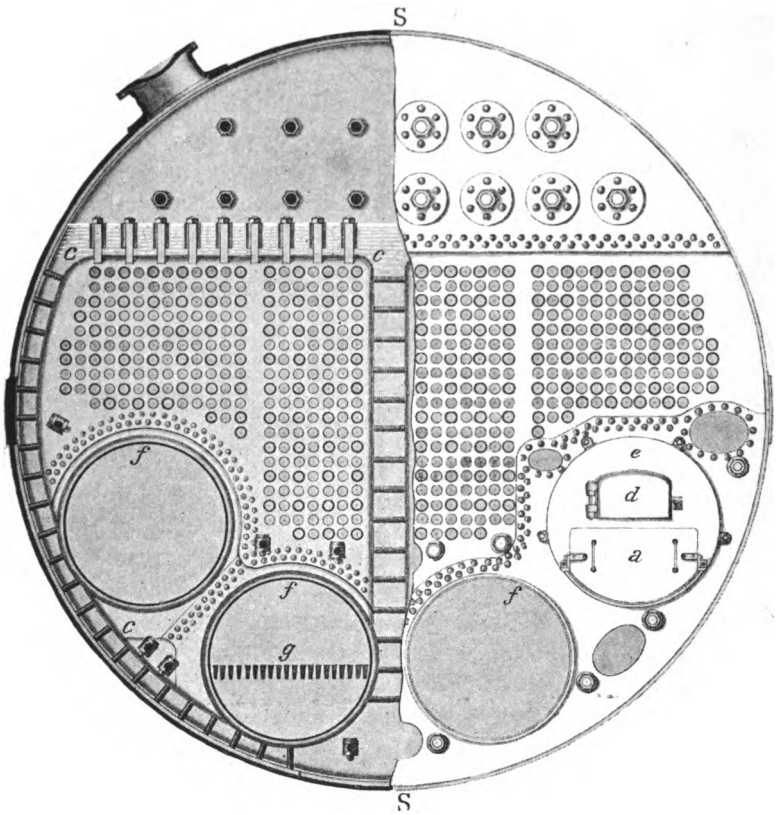


FIG. 3.

service are of two kinds, *high* and *low*, the latter being used only in some of the smaller classes of vessels.

High Cylindrical Boiler (Figs. 3 and 4)

This is the boiler in general use both inside and outside the service; its features are as follows:—

The shell, S, is of circular section, with flat ends; its diameter

being 11 to 16 feet, or occasionally more, and its length 9 to 10 feet.

The furnaces, *f*,—two, three, or four in number according to size of boiler,—are also of circular section; they are usually between 3 feet and 3 feet 9 inches in diameter, and about 6 feet 6 inches in length.

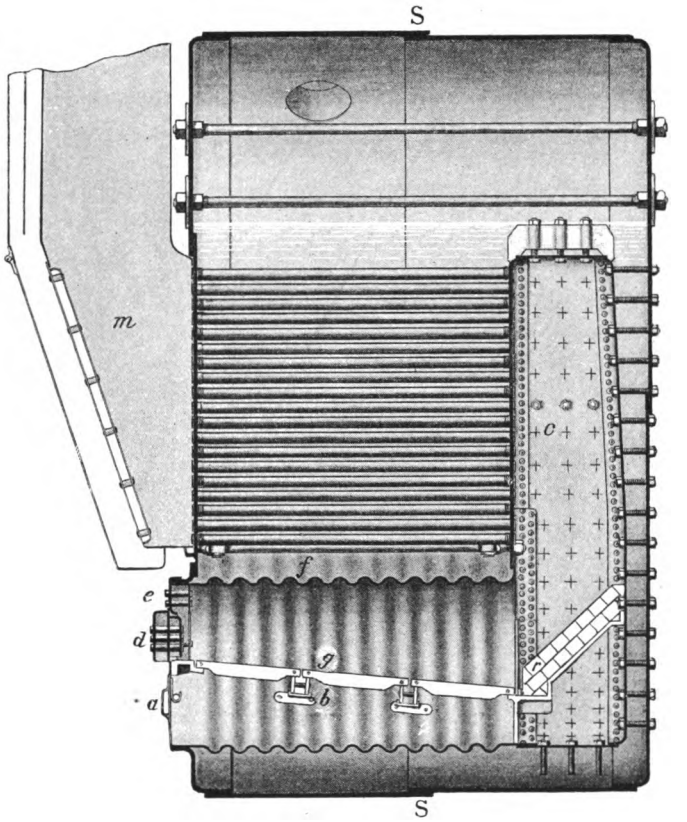


FIG. 4

The combustion-chambers, *c*, are situated near the back end-plate of the boiler, and rise to a height sufficient to allow of the tubes returning over the furnaces to the front end-plate. They are either two or three in number; *i.e.* one to each furnace in boilers with two or three furnaces, and, in four-furnace boilers, one to each side pair of furnaces, *or*, one to the central pair and one to each side furnace.

The smoke-box *m* is an external fitting. The water space between the furnaces is 4 to 5 inches deep at the smallest part, the corrugations in adjacent furnaces being alternate, to maintain this space uniformly; between the combustion-chambers the space is 5 to 6 inches, and behind the combustion-chambers, usually, 5 or 6 inches at the lower part, and 8 or 9 inches at the upper part.

Construction and Strength

Shell.—The shell is usually constructed of two rings of plate riveted together, each ring formed of two plates, as may be seen in Fig. 22A; at the ends are the flat *end-plates.* The joints, A, connecting the two plates of each ring, are the *longitudinal joints;* the joints, B, connecting the rings to each other and to the end-plates, are the *circumferential* or *ring* joints. The former are much more severely tried than the latter.

The joints used are shown in section in Fig. 5. The longitudinal joint, A, is a *treble-riveted butt-joint with double cover-plates,* that is, the edges of the plates abut, and are covered both on the inside and the outside by cover-plates, *c*, and three rows of rivets are used for each plate; while the circumferential joint, B, is a *double-riveted lap-joint,* that is, the edges of the plates overlap and two rows of rivets only are used. The longitudinal joints of the front and back rings are not in line; those of the back ring are also placed high enough to avoid the stays of the combustion-chambers, so that the ring is made of two very unequal lengths of plate.

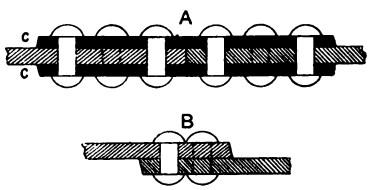


FIG. 5.

It has been stated that the pressure carried by the cylindrical shell may be increased either by increasing the thickness of the shell plates or by diminishing the diameter. It is, however, desirable to keep up the diameter of the boiler as much as possible, for this leads to economy of floor space, and admits of larger furnaces being used. The higher pressures, therefore, have been obtained by the use of thicker plates.

A limit, however, will probably be set in this direction by the difficulty of making tight riveted joints with thick plates. In

naval practice, boilers of 16 feet 3 inches diameter require a thickness of shell-plate of practically $1\frac{1}{4}''$ for 155 lbs. pressure; with the same diameter $1\frac{1}{2}''$ plates would be required for little more than 200 lbs. pressure. The latter thickness has already been slightly exceeded in certain cases in the mercantile marine. If very greatly increased pressures are to be carried in large vessels the use of smaller diameters may become a necessity; this alone would be likely to lead, sooner or later, to a change in the type of boiler.

End-Plates.—At the ends of the shell are the flat end-plates, about $\frac{3}{4}''$ to $\frac{7}{8}''$ thick. The front end-plate is in three pieces: the central piece is the front tube-plate, and the lower piece carries the furnaces, being flanged outward at the holes formed for receiving their front ends. The back-plate also is in two or three pieces; and both end-plates are flanged around the circumference for attachment to the shell. These plates require to be stayed.

Furnaces.—The furnaces, like the shell, are of circular section for strength, but are necessarily of much thinner plate ($\frac{1}{2}''$ to $\frac{5}{8}''$), since the upper part has to act as heating surface. Apart from corrosion, however, weakness in cylindrical boiler furnaces is not generally due to insufficiency of material in the plate, but to want of stiffness in the structure. The furnaces are pressed upon from *outside;* and the tendency of outside pressure upon such a structure is to increase any want of exact circularity which may exist, until finally it may cause collapse; whereas the effect of internal pressure on the same structure would be to restore the cylindrical form. Exposed as they are to great differences of temperature in different parts, the furnaces are liable to distortion, and perhaps to local overheating, and the pressure may then take effect on the slightly flattened or overheated parts. Furnaces therefore require to be stiffened to obviate the danger of collapse under high pressures.

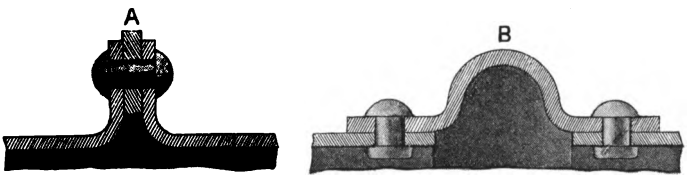


FIG. 6.

Formerly *stiffening rings* were fitted for this purpose; two forms are shown in section in Fig. 6. The plain cylindrical

furnace was made in at least two lengths; these were riveted to the stiffening ring, A or B, and the ring by its stiffness lent support to the portions of the furnace near it.

These arrangements have been superseded by the use of *corrugated* furnaces. This description of furnace is made of specially mild steel; and, after construction as a plain furnace, is rolled into a corrugated form, usually that shown in section in Fig. 4, though other forms of corrugation, shown to a larger scale in Fig. 7, are also used; with the result that the resistance to collapse under external pressure is greatly increased.

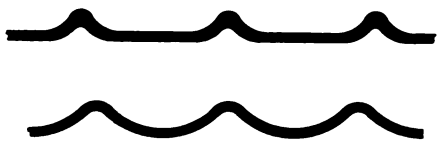


FIG. 7.

Corrugation is now universally employed; and, by its use, furnaces of large diameter, and of the thickness named above, are enabled to withstand the present high pressures with safety; with the advantage, also, that the most valuable part of the heating surface is increased in amount.

Combustion-Chambers.—The combustion-chambers are built up of flat and of curved plates, and are secured by riveting to the inner ends of the furnaces, a flange being formed on each furnace for the purpose. At the upper part this flange is turned up and secured inside the front-plate of the combustion-chamber, as in Figs. 3 and 4. The back and front plates of the chamber are flanged inwards, as in Fig. 4, to take the top and side plates. The thickness of metal used is $\frac{1}{2}''$ to $\frac{9}{16}''$. All parts of the chamber require staying.

Tubes.—These were formerly made of brass in Navy boilers; later a change was made to wrought iron in order to reduce galvanic action; now they are of mild steel. The ordinary tubes are held in place only by the tight fit of the tube ends in the tube-plates. The tube end is made a good fit in the hole of the tube-plate, and when in place is rolled or expanded by a tube-expander until a sufficiently tight grip is produced to prevent leakage. Other tubes are screwed into both tube-plates with a fine-pitched screw thread, and are then slightly rolled; these tubes, numbering about one-fourth of the whole, or more in recent boilers, act both as stays for the plates and as boiler-tubes, and are called *stay-tubes*. Both kinds of tube are shown in Fig. 8.

The plain tube, *t*, is $2\frac{1}{2}''$ external diameter and $\frac{1}{8}''$ to $\frac{1}{6}''$ thick, but is slightly enlarged in diameter at the smoke-box end; this facilitates insertion and withdrawal. The stay-tube, *s*, of the same external diameter in the body as the plain tube, but $\frac{3}{16}''$ to $\frac{1}{4}''$

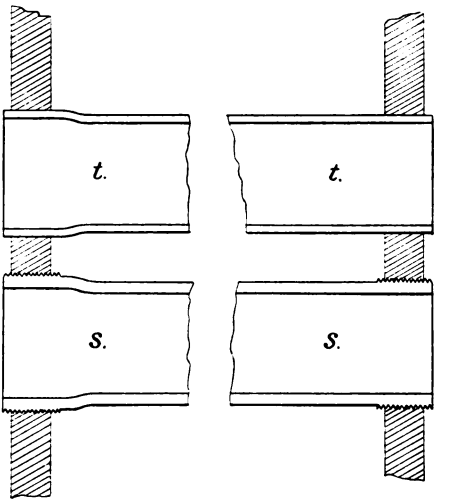


FIG. 8.

thick, is further thickened at the ends for screwing with *plus* threads. The diameters are made unequal at the two ends, to enable the smaller end to be passed clear through the hole in the outer tube-plate, but the threads are necessarily of the same pitch to admit of screwing simultaneously into the two plates.

Staying.—The parts of the boiler requiring staying are the flat (or merely curved) plates, viz. the tube-plates, the combustion-chamber plates, and the end-plates of the shell. The stays used may be seen in Figs. 3 and 4, and are shown separately in Figs. 9, 10, and 11.

Staying of Tube-Plates.—The staying of these plates by stay-tubes need not be further dwelt on, except to remark that the desirability of using these stays is sometimes questioned on the ground that they are unnecessary, and that they may prevent the plates from accommodating themselves to the requirements of the ordinary tubes, and so lead to leakage. Ordinary tubes, efficiently rolled, are no doubt able to hold the plates against the pressure

while the tubes remain tight. The general practice, however, both outside and inside the service, is to use stay-tubes.

Staying of Combustion-Chambers.—The sides and backs of combustion-chambers are stayed by short screwed stays (Fig. 9)

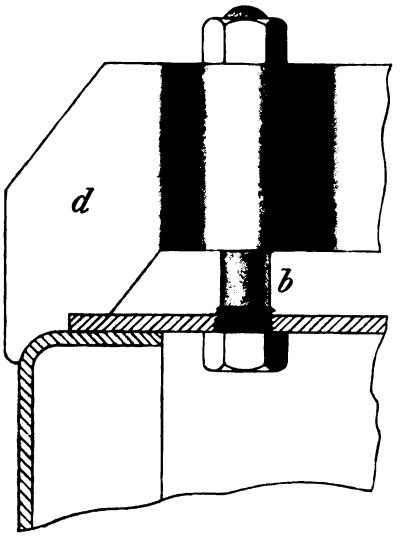


FIG. 10.

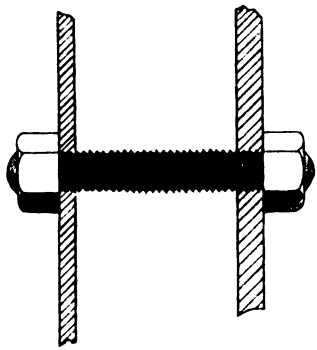


FIG. 9.

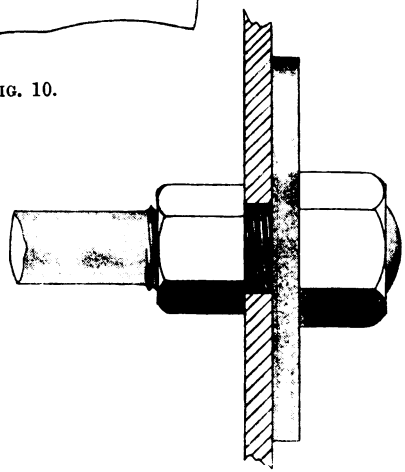


FIG. 11.

$1\frac{1}{4}''$ to $1\frac{1}{2}''$ diameter, and 6″ to 8″ apart, *screwed through* the plates, and either nutted or riveted at the ends. Nuts are used inside the combustion-chambers and outside the end-plate, and either nuts or rivet heads outside the shell.

The crowns are stayed either by direct stays from the crown of the boiler, or more usually by *dog-stays* (Fig. 10). The dog-stay is a stiff girder or dog, *d*, resting on the front and back plates of the combustion-chamber, and supporting the crown-plate by means of bolts, *b*, which are screwed through the plate, and passing through bosses in the dogs are secured by nuts. A number of such dog-stays are fitted to each chamber, spaced about 8″ apart.

The object in using these stays is to secure greater accessibility in the interior of the boiler than is possible with direct stays to the boiler shell. Where direct stays are used they are secured to T-steels worked on the inside of the shell, and are made portable, as in Fig. 14.

Staying of End-Plates.—From what has been said, it will be seen that the furnaces and stay-tubes, with the combustion-chambers and the short stays behind them, serve to tie together a considerable portion of the front and back plates of the boiler. For the rest, direct stays from end to end of the boiler are necessary. These are steel rods, 2″ or $2\frac{1}{4}$″ diameter in naval boilers, screwed at each end with a plus thread, and secured usually as in Fig. 11; the end being passed through a plain hole in the plate, and fitted with a nut and washer on the outside of the plate and a nut on the inside. The stays in the steam-chest are spaced 14″ to 16″ apart; others are fitted lower down, near the furnaces, as necessary.

Modifications.—*Oval Boiler.*—Large boiler shells were some years ago made of an oval section, the advantage being that floor space was saved for a given steam-producing capacity. The flattened sides, however, required very strong staying for even 80 or 90 lbs. pressure, and with higher pressures this form has been abandoned.

Double-ended Boiler.—The boiler hitherto described is single-ended; that is, fired from one end only. The double-ended boiler (Figs. 12 and 13) corresponds to two single-ended boilers back to back, but with a common shell, and with the back end-plates removed. Two, three, or four furnaces pass from each end towards the middle of the boiler, where the combustion-chambers are situated; and tubes return over the furnaces to the two boiler fronts. The length of the boiler is 17 or 18 feet.

The advantage obtained by this construction is economy of

weight and space; and in ordinary working the boiler has proved itself efficient. It has been for many years the boiler in general use in large vessels of the Mercantile Marine. Under forced draught, however, it has given great trouble. This has been especially the case where the opposite furnaces have been fitted

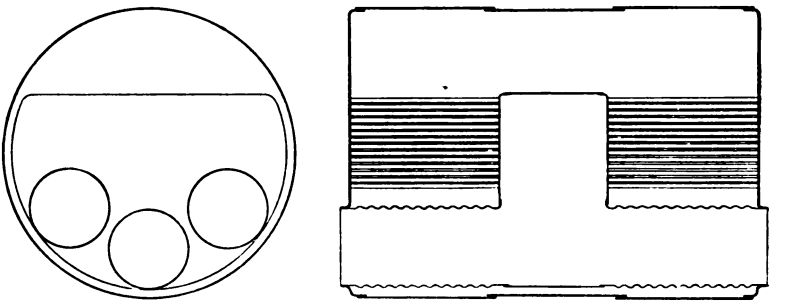


FIG. 12.

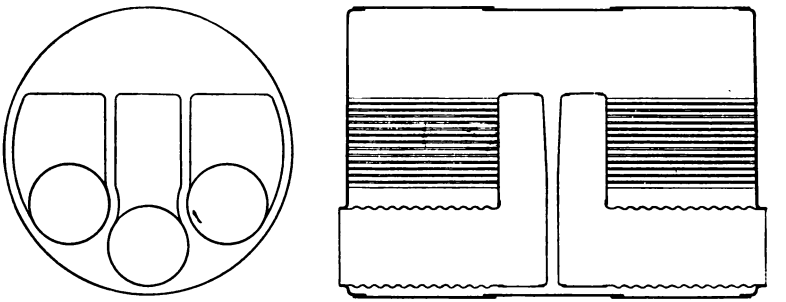


FIG. 13.

with a common combustion-chamber, or, still more, where one large chamber has been used common to all the furnaces, as in Fig. 12.

In all the later double-ended boilers, independent combustion-chambers, separated by a water space, have been used for opposite furnaces, and the furnaces at each end fitted with either two or three chambers—making, in the latter case, six chambers in all for the boiler. This is shown in Fig. 13.

Double-ended boilers are undoubtedly subject to greater strain under forced draught than single-ended boilers, and the latter have the advantage also of affording greater subdivision of the boiler power.

Low Cylindrical Boiler (Fig. 14)

This is fitted to the smaller classes of vessels, such as 3rd class cruisers and gun-vessels, where these are not provided with the special torpedo-boat type. In such vessels, the diameter of boiler that can be placed in the available depth is insufficient to allow of the return-tube arrangement being used with advantage, that is to say, with a reasonable size of furnaces and proportion of heating surface. The tubes are therefore placed beyond the combustion-chamber, and pass on to the back end-plate of the boiler; and they are kept down so that the upper row of tubes and the crown of the chamber are about on a level with the crowns of the higher furnaces. The diameter of the shell thus becomes reduced, being 7 to 9 feet with two furnaces, and about 10 feet with three furnaces. On the other hand, the length becomes increased to 17 or 18 feet; so that the boilers occupy a greatly increased length of floor space in proportion to the power, which is a great disadvantage.

The smoke-box is at the back end of the boiler. A single combustion-chamber is used; this is usually fitted, as in the example shown, with a *hanging bridge,* an arch of fire-brick supported on brackets at the sides, and built up to the roof of the combustion-chamber. The object is to deflect the furnace gases, and so cause a larger proportion of them to pass through the lower rows of tubes.

Construction and Strength.—The construction is the same as that of the high boiler, except that three rings of plate are used to make up the entire length of shell. The thickness of shell-plate for a given pressure is less, on account of the reduced diameter; $\frac{3}{4}''$ being sufficient with 10 feet diameter and 155 lbs. working pressure. The end-plates are $\frac{5}{8}''$ to $\frac{3}{4}''$ thick; other thicknesses are the same as in the high boiler. The parts stayed, and the manner of staying them, are also the same, except that direct stays, as in the figures, are more generally used than dog-stays for the crown of the combustion-chamber.

The following are some particulars relating to cylindrical boilers as used in the Navy; the numbers are approximate. The total heating surface per square foot grate is 28 square feet (high type) to 32 (low type); the tube surface per square foot grate is 24 square feet (high type) to 28 (low type); the maximum

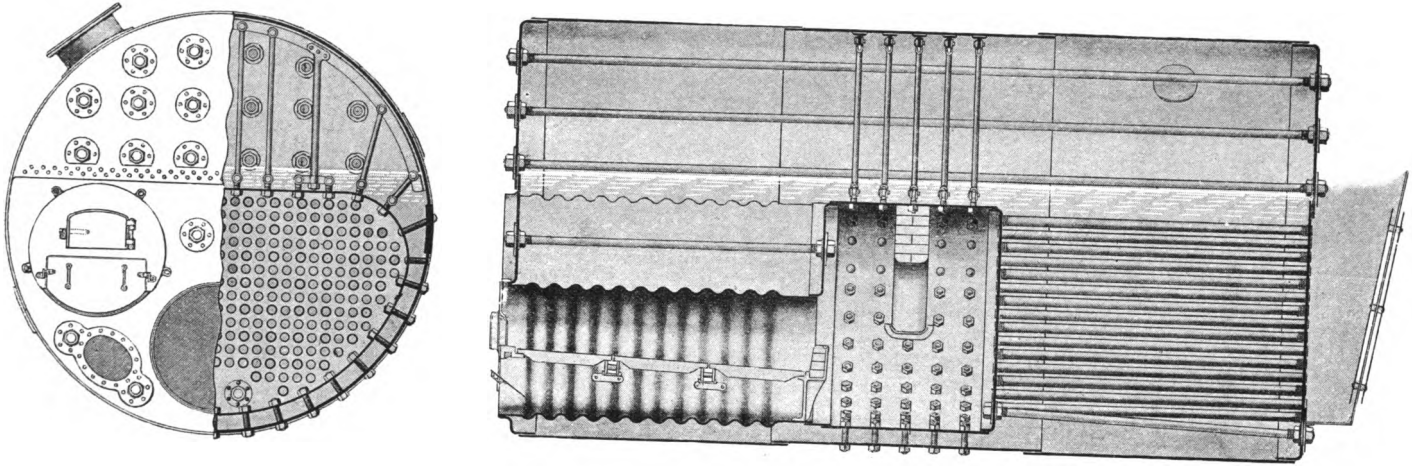


FIG. 14.

combustion per square foot grate, with natural draught, is 22 to 24 lbs. per hour; and the maximum I.H.P. per square foot grate, with natural draught and modern engines, is 11 to 12. Where the "natural draught" is produced by a small air-pressure in closed stokeholds, the rate of combustion may reach 26 lbs., and the I.H.P. 13 per square foot grate. The weight of the boilers, including water, per I.H.P., is at the natural draught power $1\frac{1}{4}$ to $1\frac{1}{3}$ cwt., and at the forced draught power less than 1 cwt.

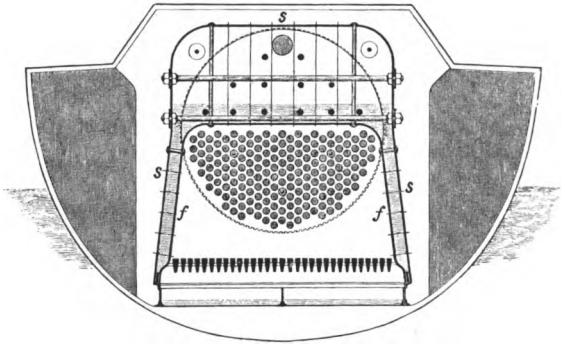
Special Types

Locomotive Boiler (Fig. 15).—This type is used where lightness and compactness in proportion to power developed are of supreme importance, as in torpedo-boats and high-speed gun-boats. The illustration represents a torpedo-boat boiler. It will be seen that the internal parts consist of a broad and deep *fire-box, f*, more or less rectangular in form, and of tubes which run from the fire-box to the back end of the boiler. The total length of large boilers of this class is 16 or 17 feet. The shell of the boiler consists of two parts: the *fire-box shell*, S, of similar form to the fire-box, and enclosing it on the sides and front; and the cylindrical *barrel*, B, enclosing the tubes. The ends are closed by flat plates. The fire-box and fire-box shell are joined at the lower part as shown, and are without bottom.

The grate extends entirely across the lower part of the fire-box, and is of large dimensions for the size of the boiler. The amount of heating surface is also large, even relatively to the grate surface. From the construction also, the amount of water carried in the boiler is exceptionally small. These special features, combined with the use of a strong forced draught, make the boiler specially small and light in proportion to its evaporative power.

In large sizes the barrel is about 6 feet in diameter, thickness of shell $\frac{1}{2}''$ to $\frac{9}{16}''$, fire-box $\frac{3}{8}''$. All parts except the barrel require staying. The fire-box and the fire-box shell are closely stayed to each other at sides, front, and crown by stays screwed through the plates, and nutted or riveted.

Forced draught is an essential element in the high steam-producing power of this boiler; and in torpedo-boats at full speed an air-pressure is used in the stokehold of 4″ to 5″. With this



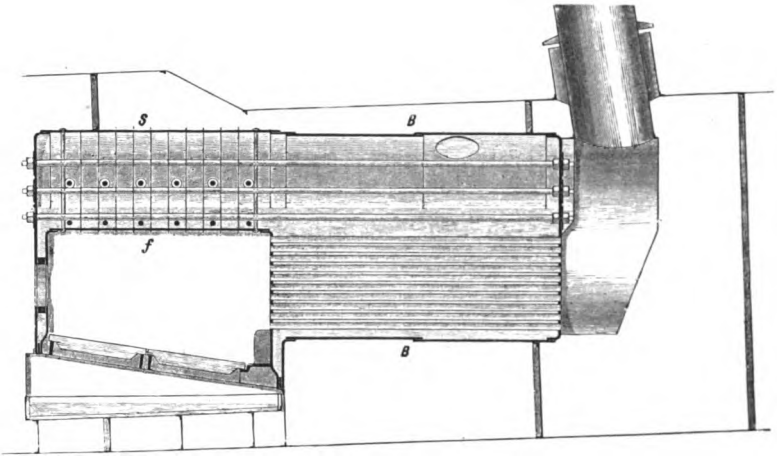


FIG. 15.

extreme draught, however, and the consequent high rate of combustion, the tubes soon become fouled or choked and the fires dirty; so that after 3 to 4 hours' steaming, full speed cannot be maintained.

The interior of the locomotive boiler is very inaccessible for cleaning; and as even a slight incrustation on the fire-box might cause disaster, it is highly desirable that fresh water only should be used. A double-distilling apparatus is carried in the large boats, as in ships, for renewing the fresh water unavoidably wasted, and for supplying the wants of the crew; and a small reserve of fresh water is also carried in a tank.

For comparison with the figures given for ordinary boilers under natural draught, it may be stated that at *full power* the rate of combustion is about 80 lbs. per hour, per square foot of grate; the I.H.P. is 25 to 30 per square foot of grate; and the weight per I.H.P., including water, is only about $\frac{4}{10}$ cwt.

In the smaller boilers of the 2nd class boats, arrangements are provided by which steam may be got up quickly in an emergency. The boat boiler can be connected with any of the ship's boilers in which there may be steam (the pressure not exceeding 60 lbs.) by a flexible hose; and by means of suitable nozzles steam may be blown in at the bottom of the boiler, which has been previously filled with water to the working level. The fire is lighted at the same time; and the fan is worked by the steam from the main boiler until there is sufficient pressure in the boat boiler to allow of its being disconnected. With this plan, and everything in readiness, steam may be got up in about a quarter of an hour, though this of course is not good for the boiler.

The use of the locomotive boiler has been extended to high-speed gunboats; and in these boilers the construction has been modified by closing the bottoms of the fire-box and the fire-box shell, having a water space between the two bottoms, and another dividing the fire-box vertically in the centre. The object is to improve the circulation, which in the ordinary type is defective from the cramped nature of the water spaces at the sides of the fire-box. These boilers are worked with a smaller air-pressure than that used in torpedo-boats, and the maximum I.H.P. per square foot grate is about 20.

Tubulous or Water-Tube Boilers.—If a cylindrical boiler shell be greatly reduced in diameter so as to become a small tube,

it may be made thin, and yet will stand a very high pressure.

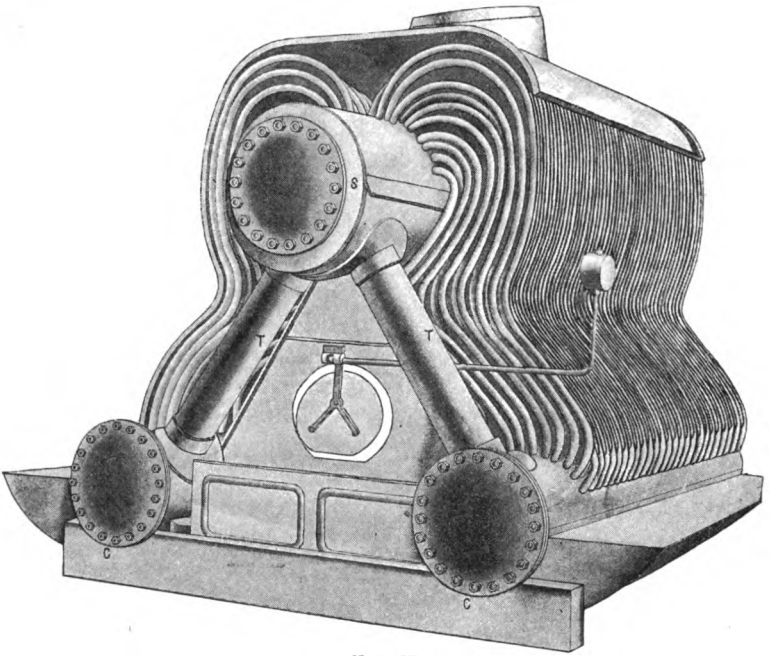


FIG. 16.

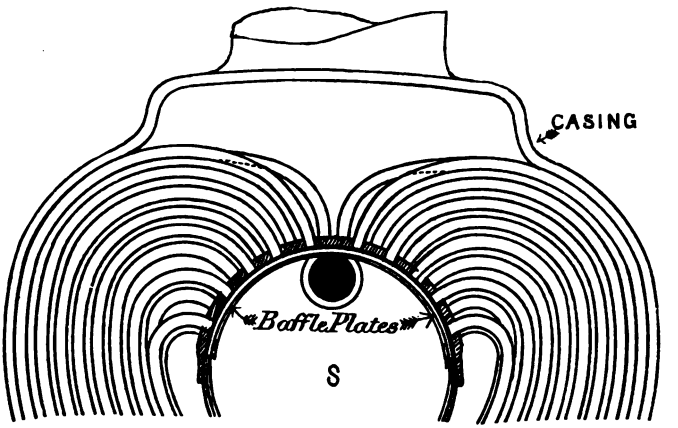


FIG. 16A.

A number of such tubes connected together, containing the water and steam, and having the heat applied to the exterior, make up

a tubulous or water-tube boiler. It appears probable that some such arrangement represents the boiler of the future.

Tubulous boilers have not yet been very extensively tried, and the earliest experience with them in ships was not encouraging. At present, however, they are as a class receiving great attention: a considerable number are at work in torpedo-boats in our own and foreign navies; and the type known as the Belleville boiler, which has been used for some years past in the vessels of the Messageries Maritime, is being fitted, in common with other types less well known, to vessels of all classes in the French Navy. The Belleville boiler is also about to be tried on an important scale in our own Navy.

The best-known marine boiler of the water-tube class in this country at present is the Thornycroft boiler (Fig. 16); this is being tried in our service in torpedo-boats and catchers, and also in a torpedo gunboat. In this boiler a great number of tubes, about $1\frac{1}{4}''$ external diameter, and curved as shown in the figure, connect a cylindrical water-chamber, C, on each side of the grate with a central steam-chamber, S, above; these chambers are further connected outside the boiler proper by two large tubes, T, at the front, which are designed to promote circulation of the water. The boiler is filled with water up to nearly the middle of the steam-chamber. When the boiler is at work, the steam and water, rising in the small tubes, are discharged into the upper part of the cylinder, *s*, against baffle plates (Fig. 16A) which cause their separation; and the water, falling and becoming mixed with the feed-water which is introduced into the lower part of the cylinder, S, descends with it by the circulating-tubes to the water-chambers. The steam is taken from the upper part of the steam-chamber by an internal steam-pipe and stop-valve as usual.

The design has recently been modified, but the characteristic features of the boiler remain the same.

The tube ends are merely expanded in place, like those of an ordinary boiler, but are not here exposed to the fire.

In the Yarrow boiler (Fig. 17), which is also undergoing trial in high-speed boats, the features are very similar; but the tubes connecting the water-chambers with the steam-chamber are straight, and discharge into it below the water-level instead of above. The chambers are constructed to be secured by joints,

as shown in the figure, so that on breaking the joints the tube ends become accessible.

In the Belleville boiler straight tubes of much larger diameter, usually 4″ to 5″, are used; these are connected and inclined to each other as in Fig. 18A to form a zigzag nest of tubes, providing a steady rise from a water-chamber (which in this boiler is of small section) to a steam-chamber. A number of such zigzags or

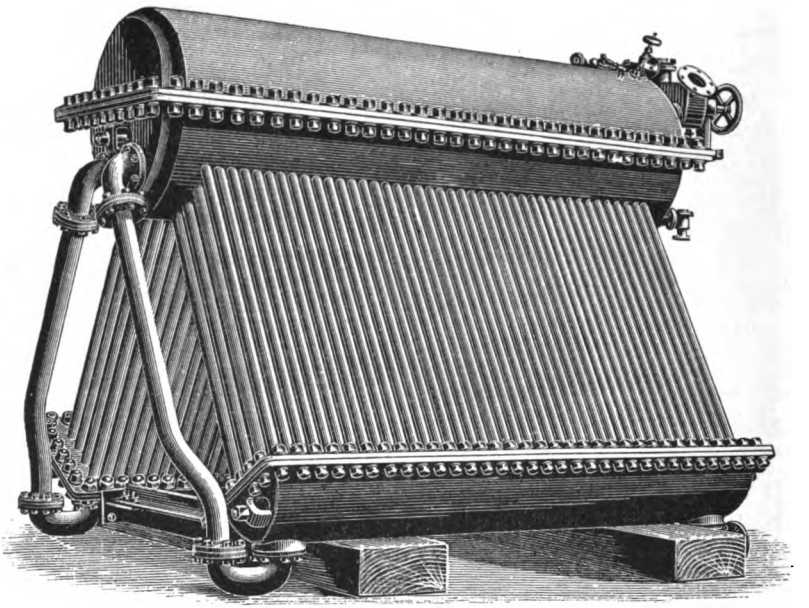


FIG. 17.

elements, *b*, *b*, Fig. 18, connecting the water-chamber or *feed-collecting tube*, *c*, with the steam-chamber, *s*, constitutes the boiler, which is set down over an ordinary fire-grate. With the exception of the steam-chamber the boiler is enclosed, as in the other types, within a non-conducting casing. Outside the casing is a circulating pipe, *p*, connecting the bottom of the steam-chamber with a sediment collector, a vessel in communication with the feed-collecting tube. The feed-water, admitted to the steam-chamber at the end opposite that where the circulating pipe is situated, becomes well mixed with the hot water discharged into the chamber by the tubes before descending with it to the sedi-

ment collector; this vessel provides a comparatively quiet space where foreign matter may become separated out before the water passes on to the feed-collecting tube, which it does from the upper part of the sediment collector.

The connections at the ends of the tubes are made by elbow-pieces or junction-boxes, *c*, *e*, to which the tubes are connected by screwed joints; and at the front ends are small doors which enable the interior of the tubes to be examined and cleaned.

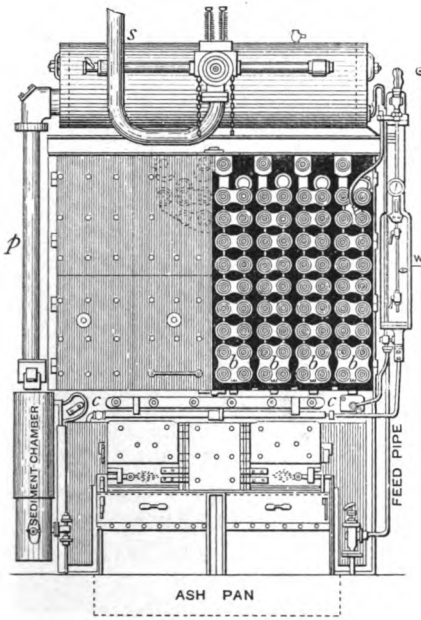


FIG. 18.

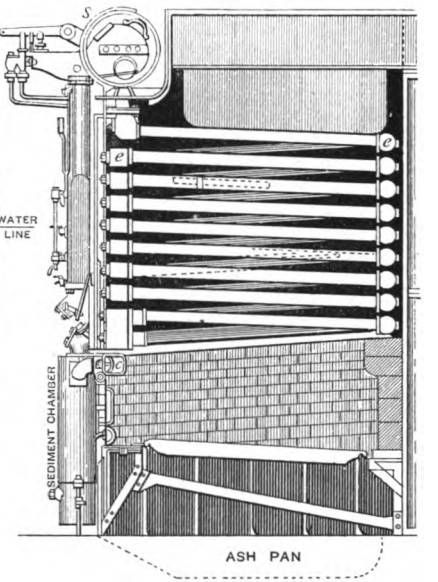


FIG. 18A.

A feature of the Belleville boiler is the automatic feed-controlling gear, which consists of a float rising and falling with the level of water in a stand-pipe connected with the boiler, the float controlling a feed-regulating valve. The rapid fluctuations of water-level and of pressure to which water-tube boilers are subject, as the result of the rapidity of evaporation and of the smallness of the body of water contained in them, are thus reduced. Steadiness in the steam-pressure at the engines is also promoted by keeping a considerably higher pressure in the boilers than is required for the engines, and reducing to the point required by means of a reducing valve.

It is especially important in tubulous boilers that the water should circulate vigorously, otherwise tubes are liable to be left for a time short of water, and to be burnt; this constituted the principal difficulty in the early forms, but appears to have been remedied in the later designs. The risk of burning is of course much greater if salt water has been used, as may have happened from leakage at the condenser tube ends. The difficulty of thoroughly cleaning or renewing the tubes is very great in most types. Non-conducting deposit, priming, and corrosion are all likely to be more serious matters with these than with ordinary boilers; and it is much more difficult to keep the water-level and the steam-pressure steady, in the absence of special fittings, from the cause before mentioned.

These boilers, however, have the great advantage that, while strong enough to carry steam of very high pressure with comparative freedom from risk of serious accident, they are extremely light in proportion to their steaming power; this arises from the smallness of the amount of water, the large grate, and the fact that the boiler consists almost entirely of heating surface. As a class, also, they are able to stand the conditions of forced-draught working. Thus a high rate of combustion may be employed; and it is claimed that steam may be got up rapidly and the draught rapidly increased or diminished without causing any trouble.

Independently of forcing, however, these boilers have so high a steam-producing power for their weight that a natural or a very moderate forced draught may well be used, with the corresponding advantage of a superior evaporative efficiency.

In the case of the Belleville boiler forced draught cannot be used except to a very moderate extent; but, from the size and accessibility of the tubes, this type has advantages for general work at sea.

Forced Draught applied to Ordinary Boilers.—The power represented by a given set of boilers depends on the amount of steam they can produce in a given time, and this largely depends on the draught to the fires. The immediate object of forced draught is to increase the rate of combustion of the coal, and consequently the rate of evaporation; the final effect being to increase the power developed with a given weight of boilers carried in the ship.

Two systems principally are at present in use: in the first, air is blown directly into the ash-pits, which are closed, the stokeholds remaining open as usual; in the second, which is the plan used in the service, air is blown into closed stokeholds. The latter, known as the *closed stokehold system*, is merely an extension of the plan adopted in torpedo-boats. All openings to or from the stokehold are closed by air-tight doors, except the inlets for air in connection with the fans, and the outlets through the fires to the funnel. A pressure slightly in excess of the atmospheric pressure is produced within the stokehold, and an increased flow of air into the fires is the result.

The fans are situated in the larger classes of vessels on a horizontal ceiling, which is worked across the stokeholds from the bunkers and bulkheads to the upper part of the boilers, or, in some cases, entirely over the boilers. When the doors in the ceiling are closed it becomes an air-tight flat.

In smaller classes the fans are fixed under the protective deck, which in these vessels forms the ceiling. This arrangement is illustrated in Figs. 19 and 20. Air-trunks, A, fitted with cowls are led down from the upper deck to supply the fans, F, the protective deck being replaced by armour bars, BB, where the air-trunks and uptakes pass. The trunk communicates on both sides of the fan with the suction orifice, S, in the centre; and when the fan is revolving air flows to this orifice, and is delivered from the circumference of the fan into the stokehold.

The pressure produced is measured by an air-pressure gauge, a glass tube of U shape containing some water. One leg of the gauge is open to the stokehold, the other is in communication with the outer air; and the difference of level in the two legs measures the excess of pressure in the stokehold over that of the atmosphere. A difference of one inch represents about $\frac{1}{25}$ lb. per square inch; two inches, $\frac{2}{25}$ lb., and so on.

The maximum air-pressure allowed on trial during the last two or three years has been 1″ to 1¼″ in cruisers, 2″ in battleships, and 3″ in gun-vessels with locomotive boilers, though the full air-pressure is now rarely necessary for obtaining the power.

Arrangements.—In many ships the boiler fronts stand out through vertical air-tight screens worked from the bearers up to the ceiling, and separating the actual stokeholds from the remainder of the boiler-room space; so that only the comparatively

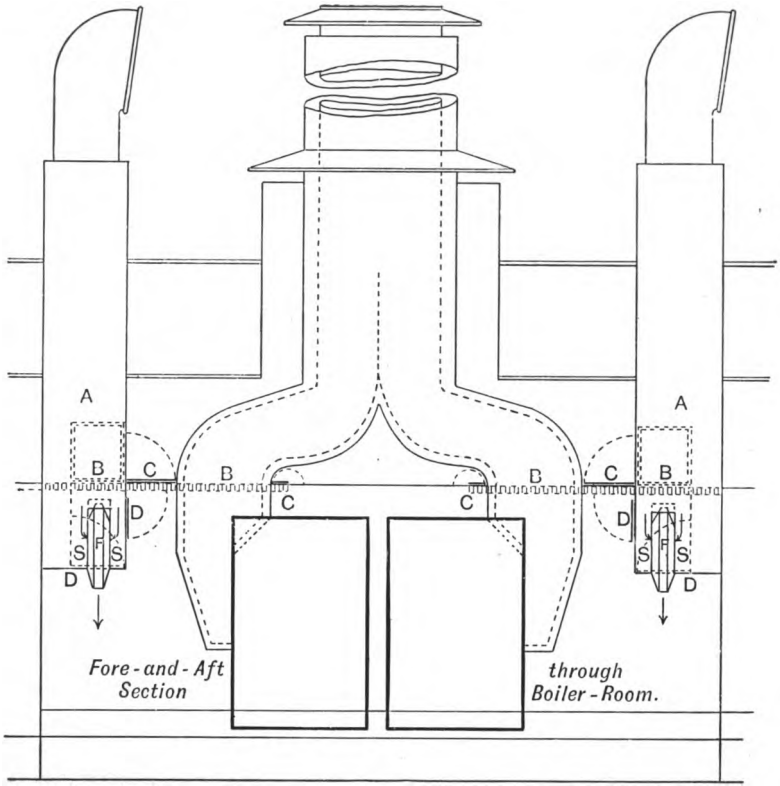

A
B
C
D
S
F
Fore-and-Aft Section
through Boiler-Room.

FIG. 19.

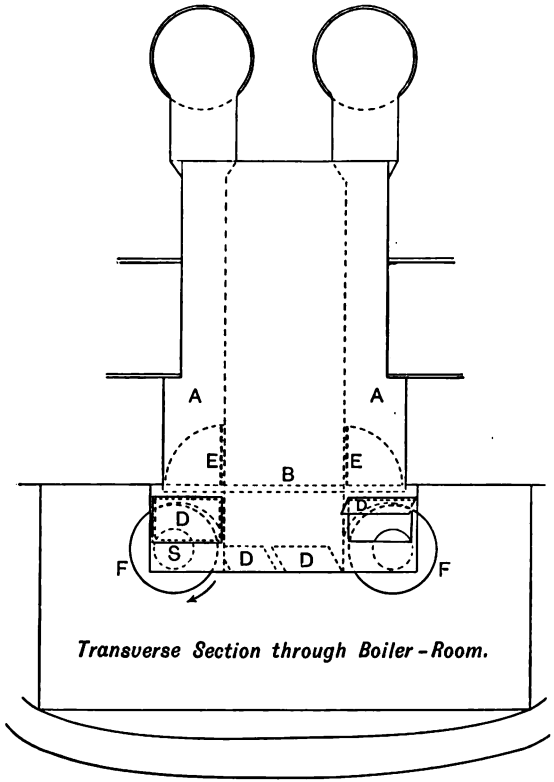

A
E
B
D
S
F
Transverse Section through Boiler-Room.

FIG. 20.

small space in the front of the boilers is under pressure. In others, as in the 1st and 2nd class cruisers, the arrangements are simplified by the omission of these screens, and the whole boiler-room is under pressure; this is the plan illustrated in Figs. 19 and 20.

Communication between the several stokeholds, or with the engine-room or deck, is made by means of *air-locks*, which are small air-tight passages fitted with two doors for entry and exit, only one of which is to be opened at a time, so that the pressure in the stokehold may not be lost. When the fans are not in use these communications are open; and doors in the air-casings and in the ceiling are also opened, to provide a downcast for natural draught, and to remove hot air from the tops of the boilers.

In Fig. 19 the doors D in the air-casings, and C in the ceiling, are closed for forced-draught working. When working with natural draught the doors D are open, as on the right-hand side of Fig. 20, and air flows naturally into the stokehold; while the doors C, being also open, allow hot air to pass from the upper parts of the stokehold.

On the left-hand side of Fig. 20 the fan is shown in operation. In many ships the openings for the natural supply of air are small, and it is frequently necessary to keep some of the fans going to assist the natural draught. An alternative plan is to close the stokeholds and use a small air-pressure, not in any case exceeding $\frac{1}{2}''$ (unless automatic ash-pit doors are used, in which case extra pressure is necessary to produce the same effect); up to this point the rate of combustion is that corresponding to a strong natural draught. The natural draught power is so obtained on trial.

Induced Draught.—A third plan of producing a forced draught consists in using fans in the uptakes to quicken the flow of the funnel gases, and so reduce the pressure in the uptakes; the draught, which depends on the difference of pressure between stokehold and uptakes, is thereby increased. Draught so produced is distinguished by the name "induced draught."

The following list will illustrate the results which have been obtained on trial with forced draught:—

Name of Ship.	Date.	I.H.P. per sq. ft. grate.	Air-Pressure.
Scout	1885	16·6	1·5
Severn	1886	15·9	2
Anson	1887	15·8	1·9

Name of Ship.	Date.	I.H.P. per sq. ft. grate.	Air-Pressure.
Australia	1887	17·3	1·75
Magicienne	1889	19·3	2·2
Nile	1890	20·1	1·75
Second Class Cruisers	1891	16	¾ to 1
First Class Cruisers	and	15	¾ to 1
Royal Sovereign	1892	18½	1·6

Assuming that 11 to 12 I.H.P. per square foot grate might be produced by natural draught, and excluding extreme cases, the increase of power obtained amounts to about 40 to 55 per cent with air pressures of 1″ to 2″, and 25 to 30 per cent with the moderate pressure of ¾″; though the rate of combustion corresponding to a given air-pressure is not very definite, but depends to some extent on the state of the fires.

With the higher air-pressures, and the very large amount of heat thereby developed, great stress is thrown on the internal parts of the boiler, especially on the inner tube-plate, a plate which can scarcely have anything but a defective circulation on the water side; and leakage at the fire-box end of the tubes is very liable to occur, more particularly when any change takes place in the air-pressure. This probably arises from differences of expansion and contraction between the tube and tube-plate. To prevent this it is highly desirable that the draught employed be neither excessive at any time nor rapidly varied.

Admiralty Ferrule.—The ferrule shown in Fig. 21 is now fitted to the tubes at the fire-box end for forced-draught working. The flange, *f*, covers the tube end and protects it from the flame, and the ferrule lines the tube, but is in contact with it only beyond the tube-plate, where there is water to receive the heat from the tube. There is thus a small air-space between the ferrule and the tube end which it has to protect. This successful fitting has enabled ships to develop their full power which, without it, were quite unable to do so; and it is now being generally applied in the service.

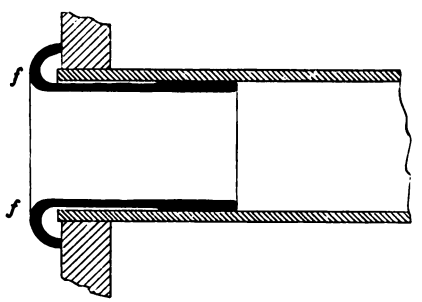


FIG. 21.

In regard to the use of forced draught, it should be remembered

that the object aimed at is a highly important one; and that, whatever may be the weight of the machinery carried, it is desirable to be able to produce from it, on an emergency, the utmost power and speed consistent with its maintenance in good working order. The forced-draught arrangements are essentially *emergency* fittings, intended to be fully used only under circumstances of great urgency. Apart from forcing, however, the fans are very useful, and in some ships necessary, for assisting or superseding the draught producible by natural means, and so rendering the steaming power less dependent on atmospheric conditions, or on peculiarities in the ship's structure.

Boiler Mountings

The principal boiler mountings are shown in position in Figs. 22 and 22A: these are, for steam,—*safety-valves, stop-valves, pressure-gauges*; and, for water,—*glass water-gauges* and *test-cocks, feed-valves, blow-out valves.* The valves and valve-boxes are all of gun-metal; the various pipes in connection with them within the boiler are of brass.

Safety-Valves (Fig. 23).—These are self-acting relief-valves, intended to prevent the pressure within the boiler from rising above a certain point, by providing a free escape for any excess of steam after this pressure has been reached. Two valves at least, sometimes three, in one valve-box are fitted to each boiler; these afford greater security than would one large valve, and also provide a larger periphery, and therefore a freer escape, for a given valve area.

The valve-box is secured over an aperture in the boiler shell, and the steam has free access to the under sides of the valves, *v, v.* The latter are kept closed against any pressure less than the blowing-off pressure by strong steel springs, *s,* outside the valve-box; these bear upon collars on the valve-spindles, and are compressed to bring the desired load upon the valves by screwing down the cap, *c.* The amount of compression is limited by shoulders on the standards, *t.* By inserting ferrules under the cap, suitable compression can be given for lower pressures.

The lift of the valves when blowing off is very small, probably not more than $\frac{1}{8}''$ at most; but a spring-loaded valve cannot lift at all without compressing its spring and thereby increasing the

load upon itself, the result being that the pressure within the boiler may increase somewhat beyond the blowing-off point.

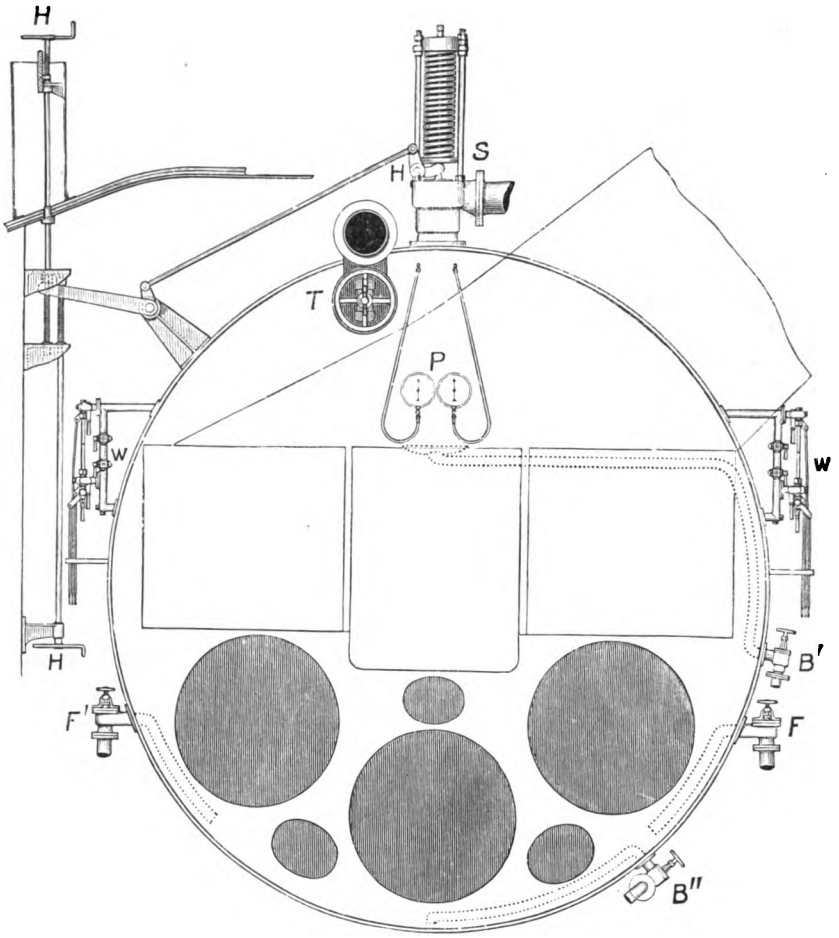


FIG. 22.

S—*Safety-Valves.*
T—*Stop-Valve.*
P—*Pressure-Gauges.*
W—*Water-Gauges.*
F—*Main Feed-Valve.*
F'—*Auxiliary Feed-Valve.*
B'—*Surface Blow-out Valve.*
B''—*Bottom Blow-out Valve.*

This effect is reduced by employing long springs, or springs of large diameter, for these are less affected by any given lift than shorter springs, or springs of less diameter. The rule for the service is that the springs used shall require a compression for

the full working load equal at least to the diameter of the valve; this implies that the small compression they will receive when the valve is blowing off will produce only a slight effect. When

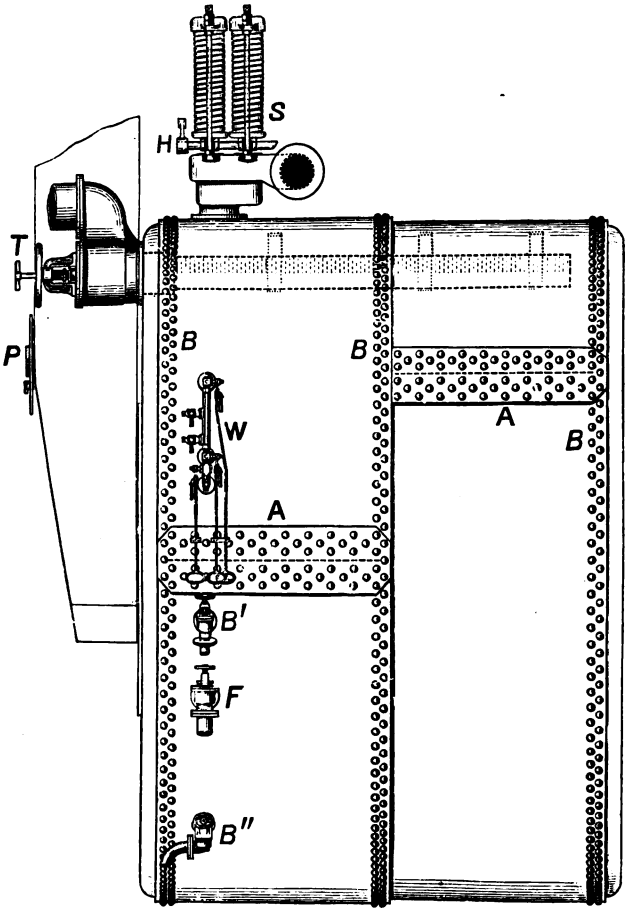


FIG. 22A.

the valves lift, steam escapes to the atmosphere by the waste steam-pipe which is carried up alongside the funnel.

Gear is fitted for lifting the safety-valves by hand at any time; this can be worked from deck, as well as from the stoke-hold. Levers *l*, moved by the gear, act on the under side of collars on the valve-spindles as in the figures, or on the under side

of the valves themselves; in either case the gear does not interfere with the lifting of the valves automatically, nor does the gear from deck interfere with that from the stokehold, or *vice versa.*

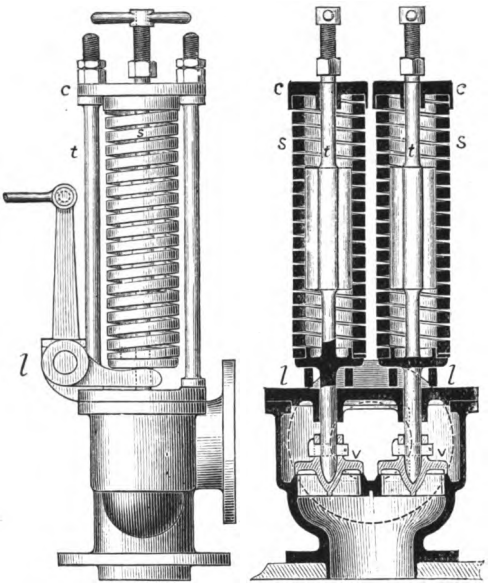


FIG. 23.

A plate recording the working and water-test pressures is fixed on the front of the boiler.

Boiler Stop-Valve (Fig. 24).—The office of this valve is to enable communication to be opened or closed at any time between the steam-chest of the boiler and the main steam-pipe. The valve-box is secured to the boiler at the upper part, and communicates with the interior by means of a large pipe, *i*, with two branches, one of which is seen in Fig. 22A, running across the steam-chest near the top; these *internal steam-pipe branches* are closed at the farther end, and take steam only through slits cut in the upper part of the pipe. The object is to check "priming," that is, the passage of water with the steam to the main steam-pipe. The tendency to this is reduced by collecting the steam from different parts by small orifices, instead of taking it at one point only by a large orifice, which would induce a violent flow in one direction.

The peculiar feature of the valve is that it is a self-closing or

non-return valve. It may be closed by hand by means of a hollow screwed spindle, S, separate from the valve, but capable of being screwed down by the hand-wheel, W, so as to bear on a shoulder, T, on the valve-spindle. When the hollow spindle is screwed back, as in the figure, the valve is left equally free to open under

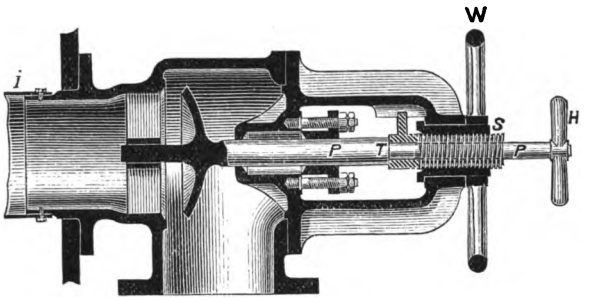


FIG. 24.

internal pressure, and to close under external pressure. The plain valve-spindle, P, passes freely through the hand-spindle, and serves with the spindle or feathers below the valve to guide the valve in its movement.

The object of the plan described is as follows:—

In the event of a boiler suffering injury in the nature of rupture, it is of the utmost importance that the injured boiler shall be instantly shut off from the main steam-pipe, so that the steam from any other boilers in connection with it may not also be discharged into the stokehold. With this fitting, if these parts remain intact, on a sudden loss of pressure occurring within the damaged boiler the valve will close automatically under the external pressure, and will isolate the boiler, thus minimising both the damage and the loss of boiler-power.

The valves are placed with their spindles horizontal, so that the weight may not affect their freedom of movement. The handle, H, on the valve-spindle allows the valve to be turned in its seat, or to be drawn out by hand when steam is not up.

Auxiliary Stop-Valve.—A second stop-valve, smaller, but otherwise precisely like the one just described, is fitted to enable communication to be made or closed between the boiler and the auxiliary steam-pipe system, from which the various auxiliary engines throughout the ship are supplied. This valve is omitted in the general view of the mountings, Figs. 22 and 22A.

Pressure-Gauges (Fig. 25).—Two of Bourdon's pressure-gauges are fitted to each boiler: one graduated to 15 or 20 lbs. above the working pressure; the other 90 lbs. higher, for use in the water test, and as a check upon the first.

A bent tube, *t*, of elliptical section, has the steam admitted at

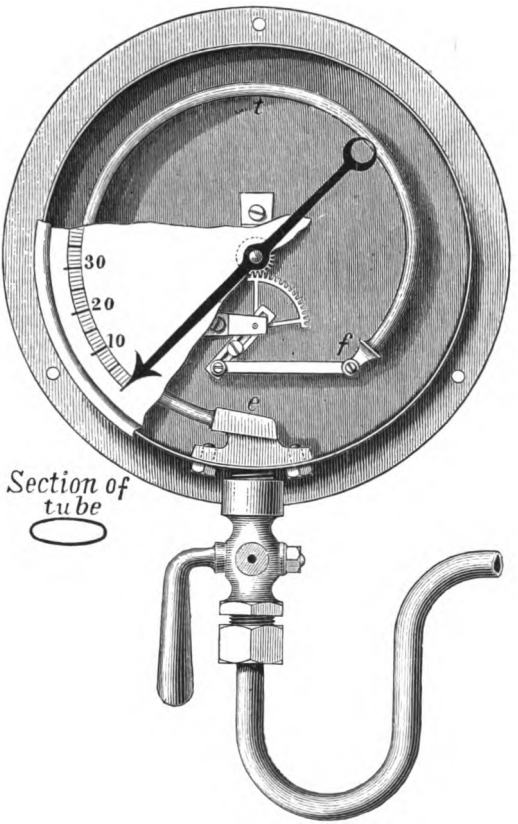


FIG. 25.

the fixed end, *e*, to its interior, while the outer end, *f*, which is free to move, is closed. Such a tube tends to straighten under internal pressure; and the resulting movement of the free end is communicated by link-work to a spindle, which carries a pointer moving on a graduated dial. The pressure shown by the gauge is not the full or *absolute* steam-pressure, but is the excess of the pressure over the atmospheric pressure in lbs. per square inch.

Feed-Valves (Fig. 26). — These are for regulating the admission of feed-water to the boilers. Two valves, precisely alike, are fitted to each boiler: one, the *main*, for regulating the ordinary supply from the main feed-pumps; the other, the *auxiliary*, connected with the auxiliary pumps. The main feed-valve is placed in service boilers on the right-hand side, the auxiliary on the left. The valves are non-return, being separate from the spindles, which are merely regulating-screws for fixing them down on their seats, or for freeing them and regulating their lift. When freed, as in the figure, the valve is lifted against the boiler-pressure by the action of the feed-pump, but closes automatically as soon as the pump is stopped, thus preventing the possible return of the water.

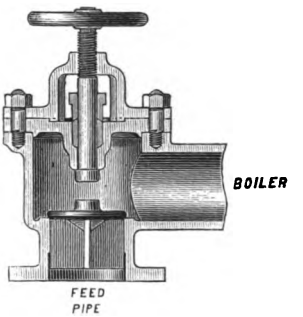


FIG. 26.

An internal feed-pipe is fitted to deliver the water clear of the furnaces, as shown in Fig. 22.

Water-Gauges (Fig. 27).—Two glass water-gauges are fitted to each boiler, for showing the level of the water. The gauge consists of a glass tube, fitted at the water-level, and placed in communication at the top with the steam space and at the bottom with the water space of the boiler, so that the water-level is visible in the glass. The ends of the glass are fitted steam and water tight into sockets, S, which are attached either directly to the boiler or to a gun-metal stand-pipe, H, attached to the boiler and freely communicating with the spaces named. Communication with the glass is opened or closed by the cocks, C; a third cock, K, allows the glass to be blown through. These cocks are opened or closed from the stokehold floor. When just visible in the glass the water stands at about 3″ (in recent cases 4″), and at half-glass at about 9″ (in recent cases 11″) above the highest part of the heating surface, the level of which is shown by an index-plate on the boiler. The screwed plugs, P, enable a wire to be used for clearing the passages at the ends of the glass.

Test-cocks, usually two, are fitted either on the stand-pipe or on the boiler; the lower one usually a little below the lowest visible point of the glass, the higher one a little above the centre

of the glass. In the event of the glass being broken or choked, these enable the water-level to be roughly tested, which is done by opening the cocks and observing whether water or steam issues from the pipes leading from them.

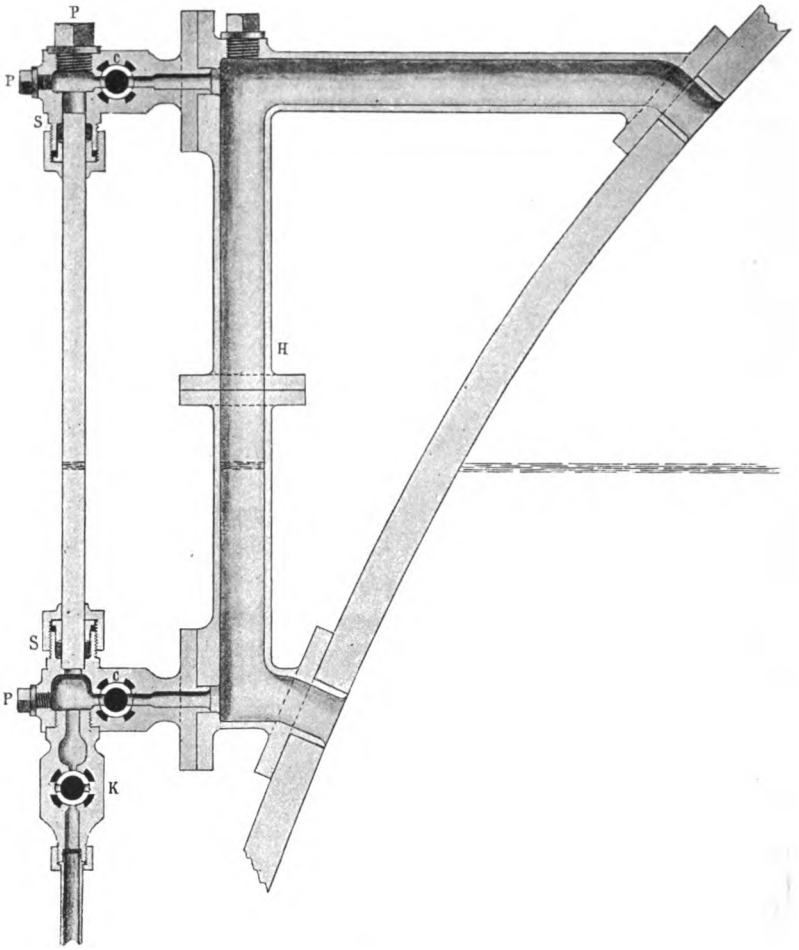


FIG. 27.

Blow-out Valves (Figs. 22 and 22A).—The object of these valves is to enable dirty water and impurities to be discharged from the boiler into the sea. The practice has been to fit two blow-out valves to each boiler: one the *surface blow-out*, the other the *bottom blow-out*. Both are ordinary screw-down valves.

The valve-boxes are secured to the shell, and have pipes communicating with the interior, and external pipes leading to the sea. The internal pipe from the surface blow-out is led to a central position above the upper row of tubes, where it terminates in an open pan just below the water-level. By opening the surface blow-out valve, the lighter impurities floating near the surface are blown out; and since the impurity now consists largely of oily matter, this is much the more important of the two fittings. The internal pipe from the bottom blow-out valve is led to the bottom of the boiler, where, formerly, the heavier impurities, such as deposits of the salts from sea water, accumulated. This valve is now of little importance in consequence of the freshness of the water; and, as it is liable to leakage, it is no longer to be found in new boilers. Its place is taken by a cock with a nozzle to which a hose can be attached; this is used for running the water into the reserve tanks when desired.

The blow-out branches from a set of boilers communicate with the sea by a common pipe, which is fitted with a sea Kingston or screw-down valve. Between each boiler and the common blow-out pipe is a guard-cock, so fitted that the spanner cannot be removed unless the cock is shut; this reduces the danger of losing the boiler-water by leakage or by oversight.

In addition to the foregoing there are minor fittings; as, for instance, an air-cock at the highest part of the shell, for use as an air-escape when filling the boilers completely with water, the hydrometer cock, and some others.

Main Steam-Pipe Connections.—When there are separate boiler-rooms, the main steam-pipes from the several sets of boilers are led independently to the engine-room bulkhead, and are there connected with a common athwartship steam-pipe, situated on the engine-room side of the bulkhead, and communicating with both sets of engines. The arrangement is represented in an elementary manner in Fig. 28, which shows four double-ended boilers placed in two stokeholds: *m*, *m* are the main steam-pipes from the two sets of boilers, *pp* is the athwartship pipe. At the junction of each of the pipes, *m*, with the athwartship pipe is a stop-valve, S, on the bulkhead, by which the set of boilers with which the particular pipe communicates may be shut off from, or put in communication with, the common pipe. The athwartship communication along *pp* is not interrupted by the opening or closing of these

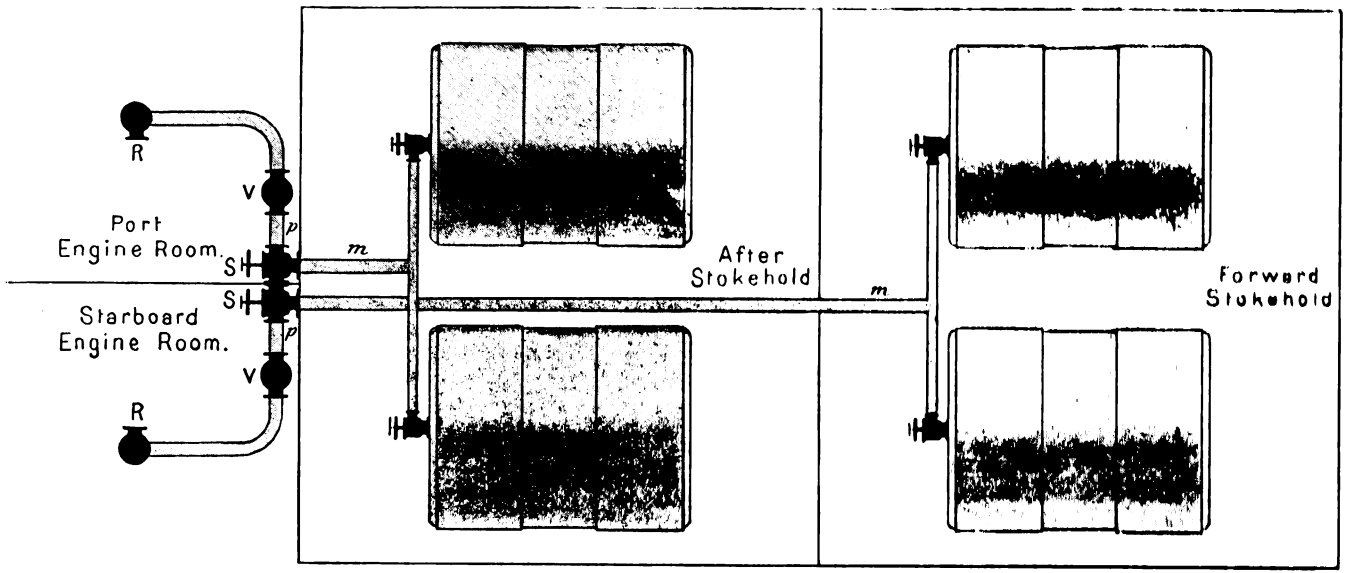

R
V
Port
Engine Room.
p
m
S
S
Starboard
Engine Room.
p
V
R
After
Stokehold
m
Forward
Stokehold

Fig. 28.

valves. The bulkhead stop-valve, S, is a self-closing valve similar to the boiler stop-valve, the object being that the valve shall close automatically in the event of such injury occurring in the boiler-room as would cause the steam in the common pipe to be discharged to it; as, for instance, the rupture of the main steam-pipe.

On each branch leading from the athwartship pipe to the engines is an ordinary screw-down stop-valve, V; this may be opened and closed by hand, either in the engine-room or from outside. The object of this valve is to enable the *set of engines* to be shut off from the athwartship pipe in the event of such injury to the engines as would cause a discharge of steam to the engine-room. The valve is fitted to open against the pressure in order that it may be easily closed, and a small pass-valve is provided to facilitate the opening of the valve when under steam.

Beyond this valve, close to the high-pressure cylinder, is the regulating-valve, R, of the engines. Another stop-valve is sometimes fitted on the athwartship pipe at the middle line bulkhead; this enables the steam to be confined to one side if desirable.

The whole arrangement described is one intended to lessen the chance of total disablement as the result of injury to the machinery, especially in action. It will be seen that, by its use, the available steam is prevented from passing to a seriously injured boiler-room or engine-room; and that, so long as any set of boilers and either set of engines are in working condition, the steam will pass to the available engines provided the athwartship pipe is uninjured.

Under Weigh

Water in Boilers.—When under weigh the most urgent matter in regard to the boilers is the maintenance of a proper water-level, since there is in ordinary boilers no automatic provision against shortness of water as there is against over-pressure. The working-level should not be lower than about 7″ above the heating surface in ordinary boilers (3 or 4 inches in the glass); it is usually kept higher, varying from one-third to one-half glass according to circumstances. The level is maintained as steadily as possible by regulation of the feed-valves and pumps; and it is desirable that the supply to all boilers be as continuous

and uniform as possible. In case of serious shortness of water, however, in any particular boiler, the supply may be concentrated on it by shutting off the others for the time; an augmented supply may be given by increasing the speed of the pump, and, if necessary, the auxiliary pump may be used also. If under these circumstances the water persistently fall and then disappear from the glasses, and cannot be made to reappear quickly, the boiler should be relieved of pressure by gradually lifting its safety-valves, and shutting its stop-valve, and by checking or drawing the fires, according to judgment; so that the surfaces, if overheated, may be exposed to as little pressure as possible.

Priming.—By this term is meant the passage of water with the steam from the boilers to the engines. The result, if the priming be heavy, is probably to leave the boilers short of water, and to subject the engines to severe shocks from the presence of water in the cylinders; and even slight priming causes waste of fuel and loss of power.

The principal causes of heavy priming are: (1) water-level being high, with rapid combustion, and violent ebullition; (2) dirty state of water, especially scum on the surface from accumulation of oily matter; (3) starting the engines at high speed, or suddenly increasing speed.

The treatment depends on the circumstances. In general the ebullition should be checked by checking the fires, the stop-valve on the priming boiler partially closed, and the engines eased if possible—the surface blow-out being freely used if it appear necessary.

Pressure.—The pressure in the boilers is kept as steady as possible by regulation of the draught and by management of the fires. There is little difficulty in this when running steadily at a given speed; but if frequent and sudden changes are made in the speed of the engines, it becomes impossible to maintain a uniform pressure. When there is reason to expect these fluctuating demands on the boilers, it is necessary to employ ample boiler-power, and to keep up the pressure in the boilers, throttling as may be necessary for the engines, in order to provide a large reserve of steam to draw upon.

If the engines are suddenly eased or stopped when working with a high pressure in the boilers and with vigorous fires, it may be impossible to prevent the pressure from reaching the blowing-

off point, especially when it is necessary to maintain the fires in a lively condition for meeting further demands. In these circumstances the *silent blow-off* enables the surplus steam to be discharged to the condensers. The fitting for each set of engines consists of a pipe with stop-valve directly connecting the main steam-pipe with the exhaust-pipe to the condenser. Its use prevents the noise of blowing-off on deck, and avoids a waste of fresh water. The valve, however, requires to be used cautiously to avoid injury to the condenser tubes.

CHAPTER III

ENGINES

Rotatory Motion

It has been stated that the ordinary steam-engine is a reciprocating and rotative engine; *i.e.* the steam is used to drive a piston forward and backward along a straight path of definite length, and this reciprocation is converted into continuous rotation, the mechanism employed being the crank and connecting-rod.

In the diagrams, Fig. 29, PG represents the piston-rod, GC the connecting-rod, and CS the crank or arm on the shaft, S; the motion is assumed to be that of a horizontal engine for convenience of explanation.

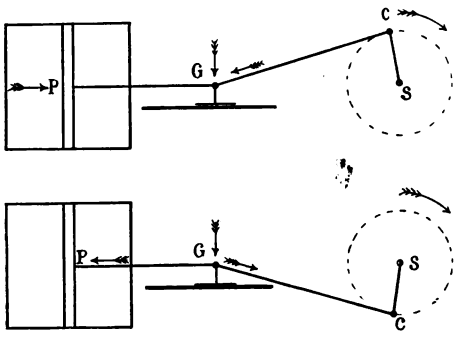


Fig. 29.

The shaft is held in its bearings so that rotation is the only motion possible to it. The connecting-rod is jointed at G and C in such a manner that it can turn freely about the joints. If the piston be driven in either direction, the crank will be turned about the axis of the shaft by the thrust or pull transmitted through PG and GC to the crank-pin; the turning force at any moment being the force acting at C tangentially to the circular path of the crank-pin, and the turning moment the same force multiplied by the length of the crank SC. The stroke of the piston is evidently twice this length.

The mechanism under consideration is both simple and effective, but there are certain peculiarities in its action which it is important to understand.

1. *Irregularity of Turning.*—It is obvious that in the two positions where the crank lies in the line of stroke there will be no turning force, whatever may be the pressure exerted on the piston; these positions are the *dead points* of the crank. Examining the state of things in other positions it will be seen that, even if the driving-force on the piston be perfectly uniform, the tangential force at the crank-pin, and consequently the turning moment, will vary from point to point of the revolution. The turning moment increases from zero at either dead point to a maximum value when the crank and connecting-rod are approximately at right angles, and decreases again gradually to zero at the opposite dead point. This irregularity of turning is inherent in the character of the mechanism; and it is of importance because it causes increased stress in the shafting, and tends to produce unevenness of rotation.

It is obviously desirable that the turning moment applied to rotate shafting should, if possible, be uniform throughout the revolution. If it vary, as in this case, a stress proportional to the greatest turning moment is brought intermittently on the shafting, while the power transmitted is proportional only to the mean moment: the stress is therefore unduly great in comparison with the power. The more irregular is the turning moment, the more severely will the shafting be strained for a given power, and the more unevenly also will the screw be revolved; whereas, for efficiency in propulsion, the screw should be revolved quite uniformly.

The above irregularity becomes more pronounced if either the steam be cut off early in the stroke for expansion, or the connecting-rod be a short one relatively to the crank.

Where a single engine is employed, the difficulty above described makes it generally necessary to use a fly-wheel, in order to carry the engine over the dead points, and reduce the unevenness of motion within tolerable limits. In marine engines, however, it has been the almost invariable practice to couple together two or three engines to drive the same line of shafting, placing their cranks, if two, at right angles, in order that the engines may be readily started ahead or astern. By this plan one engine is in

almost the best position when the other is on the dead point; and the irregularities in the total turning moment become greatly reduced, with the result that the stress also is reduced, and greater uniformity of rotation is secured.

By coupling together three engines with their cranks 120° apart, a further improvement in this respect is effected.

Summing up:—the means of securing smooth and easy turning are: to use coupled engines, to avoid an early cut-off in the cylinder, and to avoid using too short a connecting-rod.

2. *Necessity for Guide.*—Another important feature is that a guide is necessary for the outer end of the piston-rod.

Referring to the upper diagram, Fig. 29, it is evident that the force P acting along PG, and the resistance acting along CG, will together produce a downward thrust on G, tending to deflect the piston-rod; and during the entire rotation in the direction indicated the deflecting force will act downwards, as will be seen by reference to the lower diagram, which represents the lower half of the revolution.

If the engine be rotated in the opposite direction, the deflecting force at G will act upwards.

This action is resisted by fixed guiding surfaces with which the point G is connected, as will hereafter be explained.

3. *Friction at Guide and Connecting-Rod Bearings—Length of Connecting-Rod.*—If other things remain the same, the thrust or pull along the connecting-rod, and the deflecting force at G, are both greater with greater obliquity of the rod to the line of stroke, attaining their maximum values when the crank is upright. Now the maximum obliquity of the rod is greater in the case of a shorter rod than in the case of a longer one, *i.e.* shorter or longer relatively to the length of crank. Thus with a short connecting-rod there is increased pressure on the piston-rod guide and the connecting-rod end bearings, involving increased trouble and wear and tear, as well as loss of power; and, as has already been stated, a short rod conduces, for similar reasons, to irregularity of turning. It is found in practice that the minimum length desirable is $3\frac{3}{4}$ to 4 cranks; and the larger proportion is now regarded as the minimum for the connecting-rods of marine engines.

Arrangements of Working Parts

The screw engines in the Navy are of two types as regards the arrangement of their parts, viz. Direct-Acting Engines and Return-Connecting-Rod Engines. As horizontal engines, both types are in use; as vertical, the direct-acting type only.

Horizontal Screw Engines—Direct-Acting Type (Fig. 30).—In this arrangement the connecting-rod, C, is attached to the head of the piston-rod, *p*, as near the cylinder end as possible, whence it passes on to engage with the crank-pin; it therefore lies between the cylinder and the shaft. In the figure the connecting-rod and the crank are merely indicated by lines for simplicity, the detail being the same as that given under the head "Details of Working Parts" later on. The crank-shaft is carried in bearings formed in the main-bearing frames, one of which is shown at F.

Under the piston-rod head is the guide, G. These guides (one to each engine) are practically in one with the main-bearing frames, being either carried with them on a common bed-plate, or firmly bolted to them. They are also secured to the cylinder fronts, and, together with the cylinders and frames, are well supported on the bearers.

The guides, thus supported, are well placed for resisting a downward thrust; it is therefore arranged that the engines shall rotate for ahead in the direction indicated by the arrow, *i.e.* with the cranks moving away from the cylinders during the upper half of the revolution; the deflecting force is then downward during ahead motion. This result is secured by fitting a right-handed screw if the cylinders are on the port side, or a left-handed screw if they are on the starboard side. The guide is so constructed that it will also prevent the piston-rod head from being lifted by the upward thrust acting during astern motion. The detail is the same as for the vertical engine, and will be described later. A smaller guide, *g*, has usually been provided at the back end of the cylinder for the piston-rod, which in this case is prolonged to pass through the back cover. This serves, with the front guide, to relieve the cylinder to some extent of the weight of the piston.

A difficulty was experienced in single-screw ships in fitting this type horizontally, from the want of transverse space, since the working parts of this engine lie on one side of the shaft, that is, in these ships, on one side of the middle line, so that only half the width

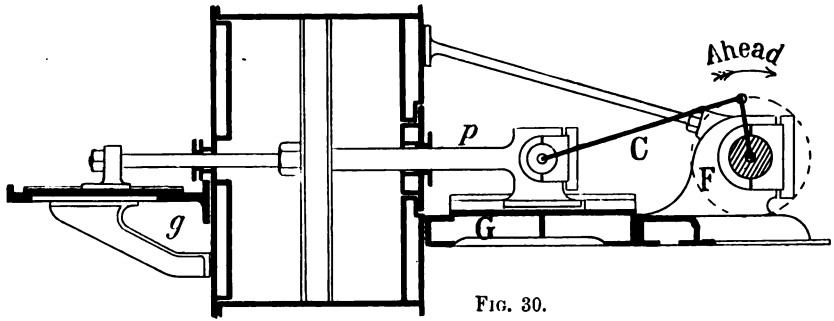


FIG. 30.

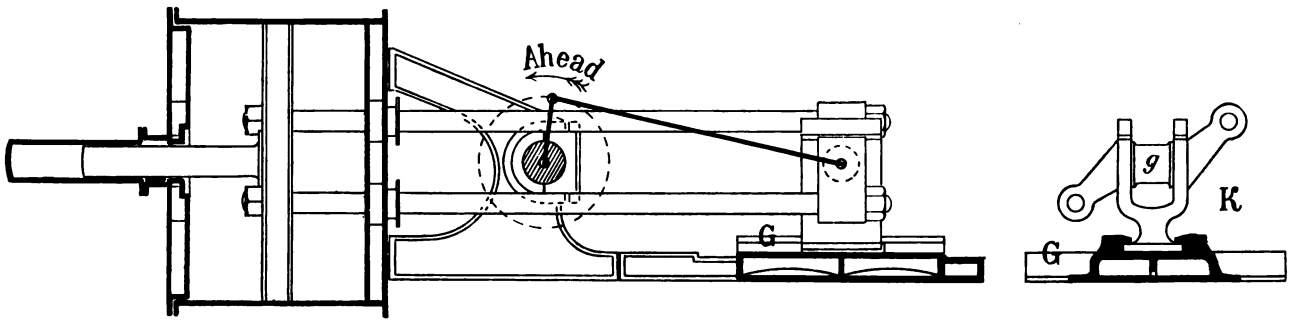


FIG. 31.

of engine-room was available. Both the stroke, and the length of connecting-rod relatively to stroke, were thus liable to become too short for good working. With regard to the stroke, a short stroke requires high revolutions in order to maintain a proper speed of piston; and though, up to a certain point, this is advantageous, the stroke must not be so short as to require excessive revolutions.

For the above reasons, horizontal engines in single-screw ships have been more generally of the return-connecting-rod type.

Return-Connecting-Rod Type (Fig. 31).—In this arrangement the difficulty above described is met by placing the connecting-rod beyond the shaft, *i.e.* on the opposite side of the shaft from the cylinder. Two piston-rods are used, which pass across clear of the shaft and of the crank-arms, and are secured to the cross-head, K. This part is shown separately in end view. It carries two arms with bosses for the attachment of the piston-rods, and also the pin or gudgeon, *g*, for the attachment of the tail end of the connecting-rod, which rod passes back from the gudgeon to the crank-pin.

The piston-rod ends require to be guided for the same reason as in the direct-acting engine; the crosshead therefore slides on a fixed guide G, similar to the one used in that engine. Since it is equally desirable in this case that the deflecting thrust during ahead-running shall act downwards, it will be seen that the engine must be rotated for ahead in the opposite direction, viz. that shown by the arrows. A left-handed screw would therefore be used with the cylinder on the port side, and *vice versa*.

This arrangement, by utilising the space on both sides of the shaft, practically increases the available space in a single-screw ship; and thus a sufficiently long stroke, and a good length of rod in proportion to stroke, are obtained.

Trunk Engine.—Another arrangement formerly largely used was that of the Trunk Engine, in which the piston-rod was replaced by a large tube or trunk; the piston being annular, and the trunk long enough to pass through the front and back ends of the cylinder. This permitted the attachment of the tail end of the connecting-rod to be made within the trunk, at the centre of the piston, instead of outside the cylinder front; with the result that space was saved, and a long connecting-rod and a sufficiently long stroke were easily obtained. For reasons which need not now be stated, this type has become obsolete.

Comparing the foregoing types, the direct-acting is the most simple and natural arrangement, and involves as few parts as are consistent with efficiency; and it would be preferred wherever sufficient length in the direction of the stroke could be given. The difficulty in fitting it horizontally applies only to single-screw ships; with twin-screws there is ample room transversely for direct-acting engines, if they are arranged as shown in Fig. 32; with the disadvantage, however, that a great length of engine-room is required. This is the ordinary arrangement of horizontal twin-screw engines.

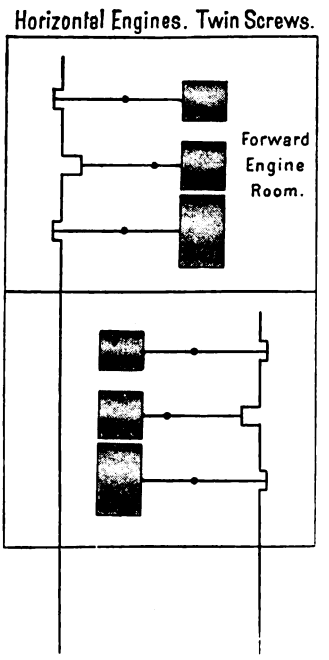


FIG. 32.

Vertical Screw Engines.—The vertical screw engine is of the direct-acting type, placed with the cylinder vertically above the shaft so that the motion of the piston is vertical. For a very long time past this arrangement has been universally employed in the mercantile marine. In the Navy the difficulty of providing protection for the machinery long prevented the use of vertical screw engines, horizontal engines having the important advantage that they may be kept entirely below the water-line. Vertical engines were, however, introduced into the service about 1871 in connection with twin-screws in vessels with side armour. It is easier to use them with twin-screws, inasmuch as the two sets of engines require less height than would be necessary for a single set of the same total power, and it therefore becomes less difficult to protect them.

The arrangement is shown in Fig. 33, its features being those already described for the direct-acting type, viz. a single piston-rod, P, the connecting-rod placed between the cylinder and shaft, and attached to the piston-rod as near the cylinder end as possible, where also a guide, G, for the piston-rod head is fitted.

The cylinder is supported by substantial upright framing to which it is bolted, consisting usually of a large back frame, F, and

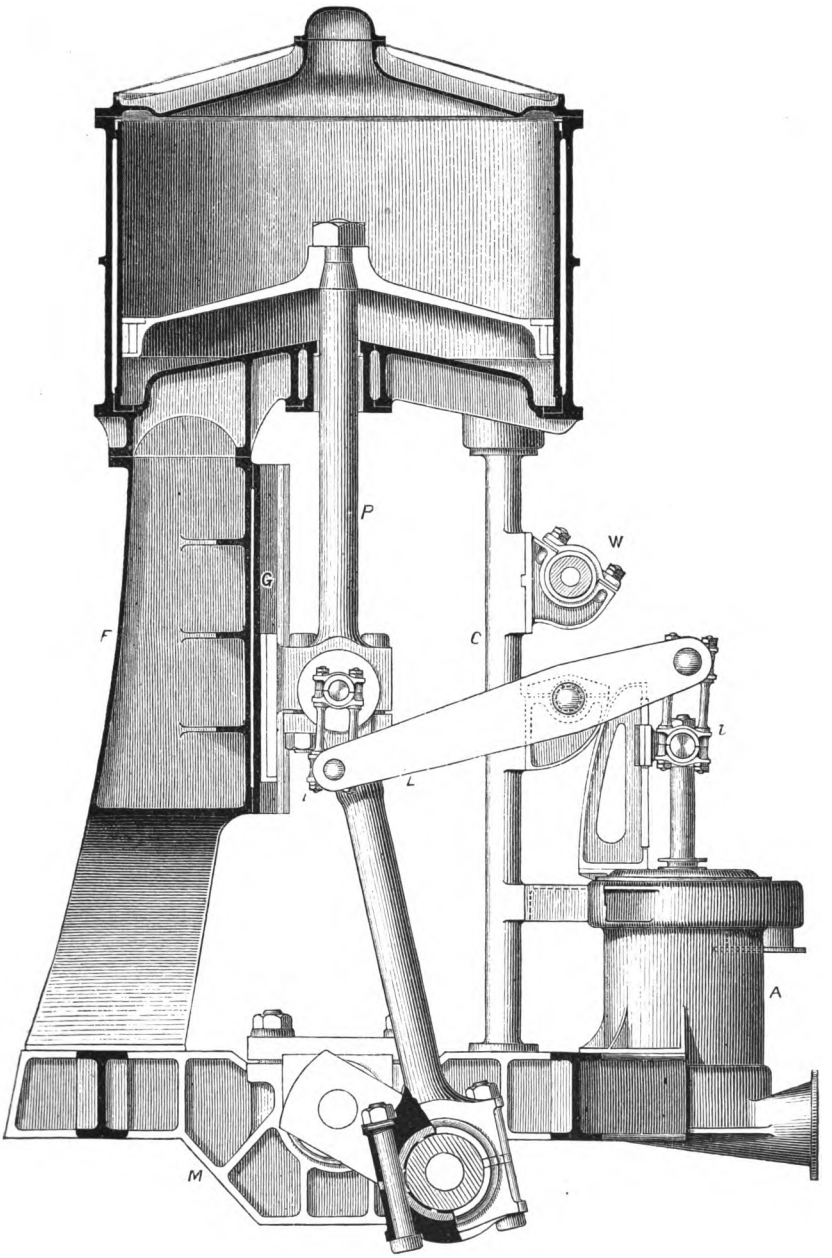


FIG. 33.

front columns, C, erected on a horizontal bottom-frame or bed-plate which is supported by the engine bearers. The guide, G, is fitted on the back frame, and is formed, as before, specially to resist a deflecting thrust in one direction, viz. *towards* the frame, though acting for the opposite direction also.

The direction of rotation for ahead, with reference to the guide, is the same as before. In short, the arrangement is simply that of the direct-acting engine placed so as to act vertically; and it combines the simplicity of this type with some important advantages arising from its vertical action. The wearing down of the

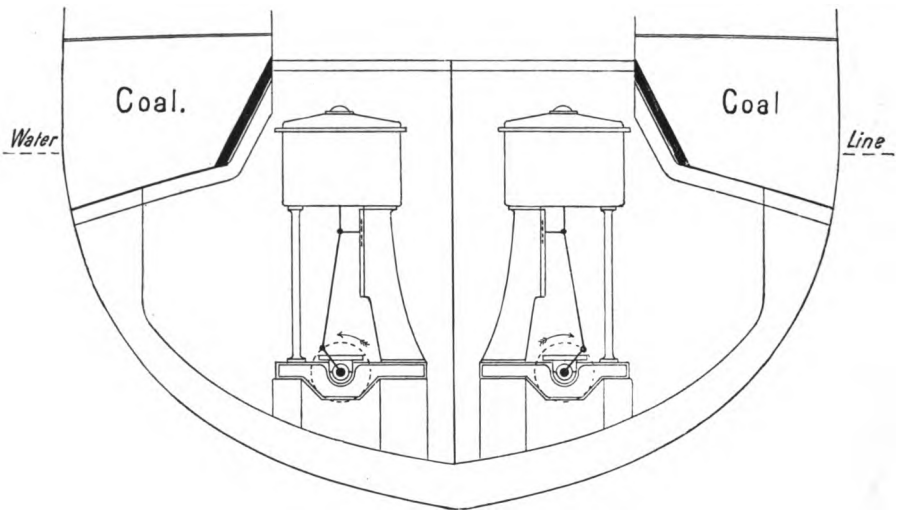


FIG. 34.

cylinders and stuffing-boxes from the weight of the piston, which constitutes a serious trouble in horizontal engines, is here avoided; the working parts are much more accessible; and, used in conjunction with twin-screws, the vertical arrangement leaves the middle line clear for a bulkhead dividing the engine-room into two watertight compartments.

These and other advantages, the value of which has been confirmed by experience, have caused this type to be so much preferred that it is now fitted to all classes of vessels, notwithstanding the difficulty in regard to protection. The length of connecting-rod used is 4 cranks; and, with a stroke suitable for

the revolutions, this brings the tops of the cylinders in most cases above the load water-line. In unarmoured vessels some protection for these parts is afforded by coal stowed abreast of the engines, and, in all but the smallest, by a protective deck; and in the larger classes of cruisers this protection is supplemented by armour surrounding the upper part of the engine-room where the cylinder tops are situated. The openings over the engines are fitted with armour bars. These arrangements for protection are shown in Fig. 34.

In recent war-ships (1890-94) the arrangement of the machinery is as follows: two sets of vertical engines in separate engine-rooms abreast, each set consisting of three engines working together on one line of shafting, and using steam on the triple-expansion system. The engine-rooms are separated by a water-tight bulkhead, or, in some cases, by a narrow compartment, as in Fig. 52. The direction of rotation of the screws for ahead is such that the upper blades turn outward; in other words, the starboard screw is right-handed, and the port screw left-handed; and the engines are set with the back frames and guides on the middle line side of the engine-room, as in Fig. 34, so that they may rotate the screws as stated when running ahead.

Paddle Engines

Two types are used: the Oscillating, and the Inclined Direct-Acting. *Oscillating Engine.*—The difficulty arising from want of vertical space is here met by the omission of the connecting-rod. The piston-rod head is suitably fitted for direct connection with the crank-pin; and the cylinder is suspended by trunnions in bearings, so as to be free to follow the movement of the crank-pin. This arrangement is an excellent one for the comparatively low revolutions required in paddle engines; and the majority of such engines are of this type. For the higher revolutions necessary for the screw it is not suitable. *Inclined Engine.*—Paddle engines are in many cases of the direct-acting type, placed at a suitable inclination to obtain the necessary length for the stroke and the connecting-rod: these are known as "inclined" or "diagonal" engines. Occasionally, in high-powered paddle-steamers, even the oscillating engine is inclined for the same reason.

CHAPTER IV

DETAILS OF WORKING PARTS

Cylinder.—The steam-cylinder is a structure with a cylindrical bore along which the piston is to be driven. The usual construction may be seen in Fig. 35, which shows a section through an ordinary marine-engine cylinder. AA is the working cylinder along which the piston moves; this part is enclosed within a cylindrical shell, BB, which stands clear of the inner cylinder except at its ends, leaving a space 1″ to 1½″ in depth around its circumference. This space is intended to be supplied with steam, and thus to form a steam-jacket for the working cylinder.

The latter part is made separately from the rest of the structure, and is secured in place as will be afterwards explained; it is known as the cylinder *liner*, while the casting which encloses it is the cylinder casting or *shell*, the whole constituting the cylinder. The ends of the cylinder casting are closed by the cylinder *cover* and the cylinder *bottom*, of which the bottom, D, at the end towards the shaft, is part of the general casting, while the cover, C, is removable. Both are strengthened by external radial ribs, as shown in the figure. Doors, *d*, in the cover and bottom allow examinations to be made, while the removal of the cover allows the piston to be taken out. The cover was formerly of cast iron, it is now cast in steel in order to combine strength with lightness.

The cylinder shell is of cast iron, being of too complicated a character to be cast in steel. It carries at one end the bottom, with feet for the support of the cylinder, and on one side a part in which are formed the passages, S, by which the steam passes to and from the interior of the cylinder. This part terminates in

the cylinder *face*, *f*, a large flat face on which the slide-valve works. The actual working face, *f*, is a false face of hard, close-grained cast iron secured to the casting, the description of cast iron suitable for the intricate casting being too soft to withstand the friction. In the cylinder face are the steam-ports or openings, *p*, which communicate by the steam-passages, S, with the ends of the cylinder. In the centre of the face is the exhaust-port, *e*, communicating with the exhaust-pipe by the exhaust-passage E.

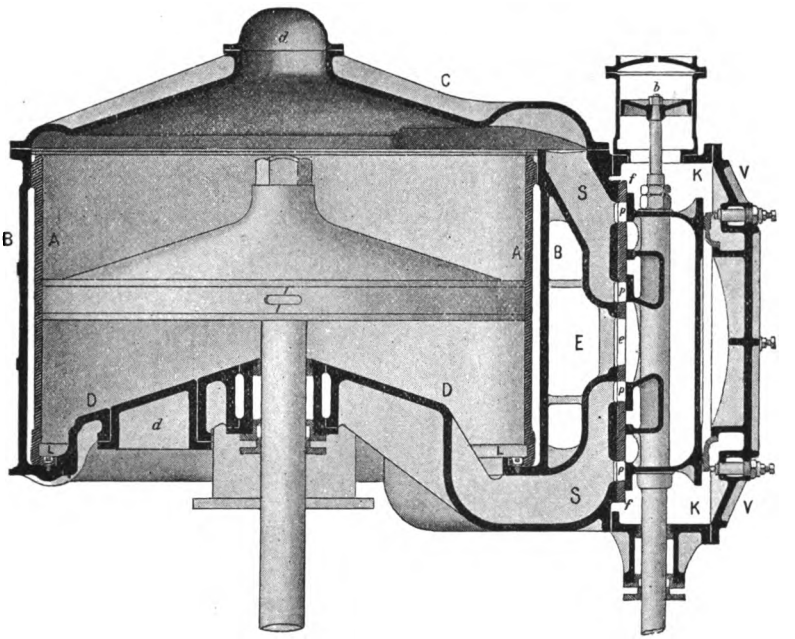


FIG. 35.

The slide-casing, K, secured to the cylinder face, or, frequently, cast with the cylinder shell, and having a removable cover, V, encloses the slide-valve, and receives the steam which is to be supplied to the cylinder. If the steam-passage at either end be put in communication with the slide-casing or with the exhaust-passage, as the case may be, steam enters or leaves the cylinder at that end accordingly. These communications are controlled by the slide-valve, the action of which will be described later.

The cylinder-liner is a simple tube or bush made of hard, close-grained cast iron, or of forged steel. The latter material

has been largely used in the Navy, especially for high-pressure cylinder-liners, but the use of cast iron liners is now general. The liner has an internal flange, LL, at one end, by which it is secured to the cylinder bottom, and an external circumferential rib at the other end, which fits tightly into the cylinder shell. The flange, L, is recessed into the cylinder bottom and secured by screws, a steam-tight joint being made. At the outer end, also, steam-tightness must be secured in order to prevent leakage from the jacket: the sketches, Fig. 36, show two methods used.

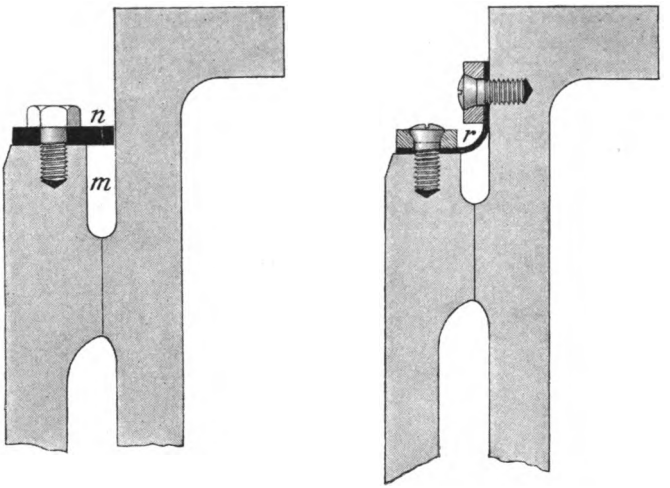


FIG. 36.

In the first a small recess, *m*, beyond the rib is tightly packed with asbestos or other packing, which is then held in place by a ring, *n*, secured to the end of the liner; in the second, which was introduced for high-pressure cylinders, a flexible joint is made between the end of the liner and the cylinder casting by means of a copper ring of the diameter of the cylinder and of the section shown at *r*. The two flanges of the ring are secured respectively to the parts named by screws, the joints being made tight by means of stiffening rings as shown; and the copper ring is flexible enough to admit of the relative movement arising from expansion or contraction of the liner. This method is becoming general for all cylinders.

The construction of cylinder above described provides a steam-

jacket space, while securing the following advantages: the liner is free to lengthen or shorten slightly under changes of temperature, independently of the shell, which freedom promotes security in working; a hard and strong material may be used for the liner, while a softer material, better suited for a complicated casting, is used for the rest of the structure; and, if defective, the liner may be easily renewed.

Cylinder Fittings.—In addition to the slide-valve, piston, and stuffing-box, which will be separately described hereafter, cylinders are provided with various fittings, as follows (these are indicated in position in Figs. 90 and 91):—

1. *Escape Valves*, E.—These are spring-loaded self-acting relief-valves in communication with the ends of the cylinder, and intended to act in the event of the passage of water to the cylinder in large quantity while the engines are at work. The impact of the piston upon the water when nearing the end of the stroke, which is liable to cause severe shock, forces the valves open and discharges the water to a pipe leading to the bilge. The springs are adjusted so that the valves will not open under the steam-pressure.

2. *Drain-Valves, d.*—These are simple valves used for draining water from the bottom end of the cylinder, the slide-casing, and the steam-jacket. They are opened and closed by hand, and are used according to judgment at any time. The cylinder drain is especially useful for clearing the cylinder before starting. The jacket drain-pipe now leads to a small receiving-vessel placed at a lower level than the cylinder and fitted with a gauge glass, so that the quantity of water present may be seen and the drain-valve used accordingly.

3. *Jacket Stop-Valve*, J.—This is an ordinary stop-valve, opened and closed by hand, which admits steam to the jacket, and is left open during the working of the engines. Further particulars are given in connection with triple-expansion engines, Chap. IX.

4. *Indicator Pipe*, I.—This is a small pipe communicating with both ends of the cylinder, and fitted with a cock with nozzle suitable for the attachment of the indicator.

5. *Starting Valve.*—Some cylinders are fitted with a small auxiliary-slide or starting valve, which can be moved by hand along a face with ports communicating with the ends of the

cylinder, so that steam may be admitted by hand above or below the piston as may be required in starting. The starting valves more usually fitted will be referred to later on.

Other fittings used in connection with the cylinders of a set of compound or triple-expansion engines will be more appropriately considered in connection with the working of these engines.

Piston.—The requirements of a steam-piston are that it shall be steam-tight in the cylinder, and yet shall move with as little friction as possible; it must also be quite unyielding, and the rubbing material must work well with the material of the cylinder.

The piston is a disc or plug, packed at its circumference with metallic packing, which consists, usually, of a single broad ring pressed out against the cylinder by springs. In Figs. 37 and 38 are shown the elevation and plan of a piston of ordinary form for a vertical cylinder, both views being partly in section. Fig. 39 is an enlarged view of the section in Fig. 37. B is the main structure or piston-*body*, and *p* is the *packing-ring*; a second ring *j*, the *junk-ring*, holds the packing-ring in place.

Details of Construction—Body.—Formerly the piston-body was of cast iron, cast in the hollow or box form shown in Fig. 42, and strengthened by internal ribs. With cast iron this form was well adapted for securing the necessary strength and stiffness with the minimum weight of the material used. Pistons are now made of cast steel; and, on account of the superior strength and reliability of this material, are cast in a single thickness, but require to be dished or coned in form, as shown in Fig. 37, to ensure the necessary stiffness. By this change in construction a considerable saving in weight is effected.

The construction at the circumference is the same in both cases. At this part the body is made deep, and carries on one of the faces a flange, *f*, while on the opposite face is the seat for the junk-ring, the latter forming, when in place, a corresponding flange. Between these opposite flanges is the packing-ring, the edges of which have a steam-tight bearing with both.

The flange and junk-ring are of a slightly smaller diameter than the cylinder, so that the packing-ring alone bears on the cylinder. At the centre of the piston-body is formed a boss, *b*, for connection with the piston-rod.

Packing-Ring and Springs.—The packing-ring is generally

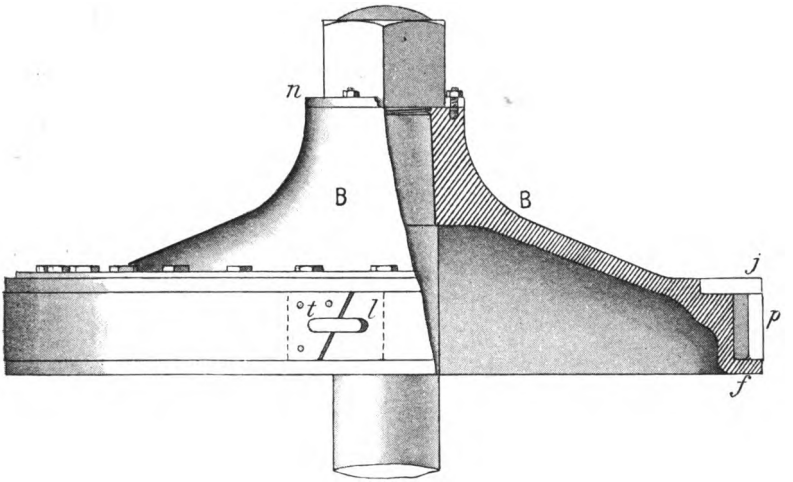


FIG. 37.

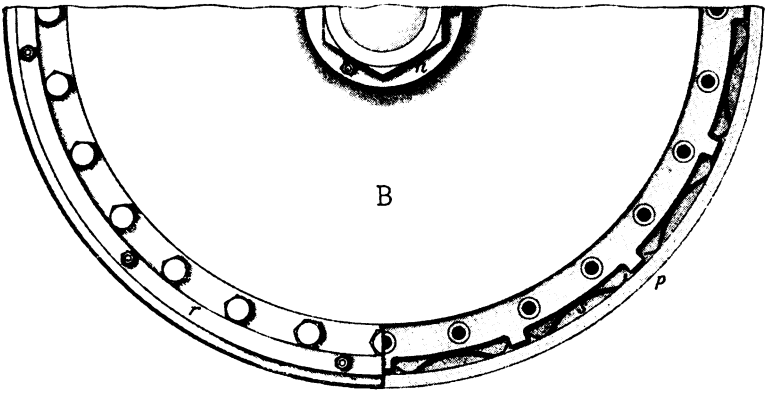


FIG. 38.

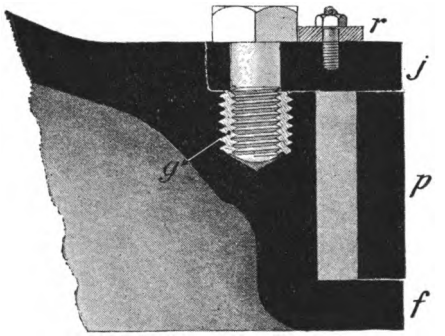


FIG. 39.

of hard, close-grained cast iron; this material works exceedingly well with cast iron liners, but with steel liners is subject to rapid wear; phosphor bronze rings are in this case preferable. The ring is cast an entire hoop, and, being turned to the correct diameter, is cut through at one part; thus freed, it is pressed radially against the surface of the cylinder by springs (*s*, Fig. 38) which are placed behind it for the purpose. Sometimes the ring is turned to a slightly larger diameter; a small piece is then cut out, and the ring is closed to fit the cylinder, springs being used as before. To prevent leakage at the part where the ring is cut a tongue, *t*, carried by one end of the ring, passes across the gap, fitting into a slot, *l*, cut in the other end; and a plate (*a*, Fig. 40), in one piece with the tongue, is fitted to the inside of the ring, closing the gap on the inside while the cylinder closes it on the outside. The tongue and cover-plate being secured to one end only, the ring is left free to adjust itself to the cylinder.

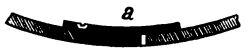


FIG. 40.

The springs used in naval pistons have generally been of the *coach-spring* pattern shown. These are spaced at regular intervals all around the piston between the ring and the piston-body; and being put into place under strain, they exert pressure on the ring, tending to force it outwards. In horizontal cylinders springs are fitted for about three-quarters of the circumference only, the space at the lower part being filled in solid, so as to ensure that the piston is supported on the broad surface of the packing-ring.

With coach-springs the pressure exerted by the several springs is liable to be unequal, and great care is required to secure uniformity in this respect. Compressed spiral springs (Fig. 41) are preferred by some makers; with these it is easier to ascertain and to regulate the pressure produced by the springs, and the pressure is less affected by slight wear in the cylinders.

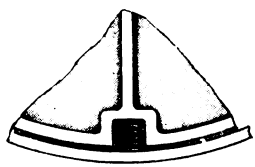


FIG. 41.

Another plan is to use a *continuous* spring, of which there are various forms; this is a single coiled spring large enough to encircle the piston-body and press upon the entire surface of the packing-ring.

Junk-Ring.—This is a stout ring of cast or forged steel

secured to the piston-body, usually by bolts, sometimes by studs and nuts. It is so fitted that when down on its seat it bears on the packing-ring, the edges of which have been scraped to a steam-tight bearing with the surfaces of the junk-ring and flange.

The junk-ring bolts, having to be frequently withdrawn, are not screwed directly into the metal of the piston, but into plugs of gun-metal (*g*, Fig. 39) fixed in the piston-body; this obviates the danger of the bolts becoming set fast in place, and if the thread should become defective these plugs can be easily renewed. The bolt-heads are frequently sunk into the ring, leaving the latter flush; but sometimes stand above it, as shown in the figure.

Guard-Ring.—To secure the junk-ring bolts from the possibility of slacking back, and perhaps causing a serious accident, a light ring, *r*, is fitted on the junk-ring, bearing against the heads of all the bolts so that they cannot turn. The guard-ring is itself well secured by studs and nuts, the latter secured by split pins.

Ramsbottom Rings.—In small cylinders, and especially in high-speed engines, the piston is frequently packed in a very simple manner by *Ramsbottom rings*. The ordinary broad packing-ring is replaced by two or three narrow rings, which, having had

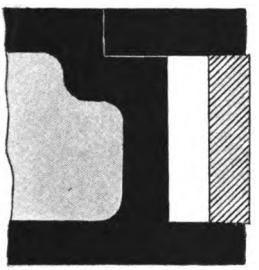


FIG. 42.

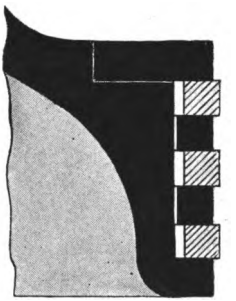


FIG. 43.

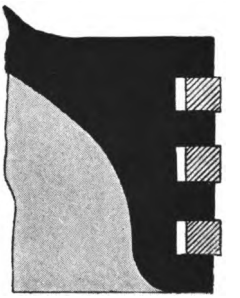


FIG. 44.

a piece cut out of the circumference, are sprung to a smaller diameter to fit the cylinder, so that they naturally tend to open. These are carried in grooves in the circumference of the piston, as in Fig. 44; no springs are used, and no junk-ring is needed.

This simple plan, which is adapted from locomotive practice, is fairly efficient in small pistons running at high revolutions, but at lower speeds is insufficient to prevent leakage. It is nevertheless frequently used for the high-pressure cylinders of triple-

expansion engines, where a small amount of leakage is of comparatively little consequence. It is desirable, however, for work at sea, to be able to withdraw the rings without removing the piston, which cannot be done with the arrangement as described. For this purpose a junk-ring is fitted, and the Ramsbottom rings are carried between separate rings acting as distance pieces, so that the whole is accessible and removable; this is shown in an elementary manner in Fig. 43.

Piston-Rod.—The piston-rod is of forged steel; it is usually secured to the piston in the manner shown in Fig. 37. A cone of small taper is formed on the rod, leaving a slight shoulder at the large end; the cone is carefully fitted to a similarly coned hole through the boss of the piston, and the rod beyond the small end of the cone is screwed, so that a nut may be used to tighten and hold the piston against the cone. This nut is kept from working loose by some means, usually a guard-plate, *n*, secured to the piston.

Stuffing-Boxes.—The piston-rod must move through the cylinder bottom as freely as possible without producing leakage. The same requirement exists in other cases where rods enter or leave vessels which are under fluid pressure, as, for instance, the slide-valve rod, pump-rods, and valve-spindles; and the same provision is made in all, though the details vary. The rod passes through a *stuffing-box* (B, Fig. 45), the aperture in which is of larger diameter than the rod, except at the bottom. Here a bush, *b*, of gun-metal is fitted, unless the entire box is of gun-metal, as in many valve-boxes and pumps; in either case, the aperture is just large enough for the passage of the rod.

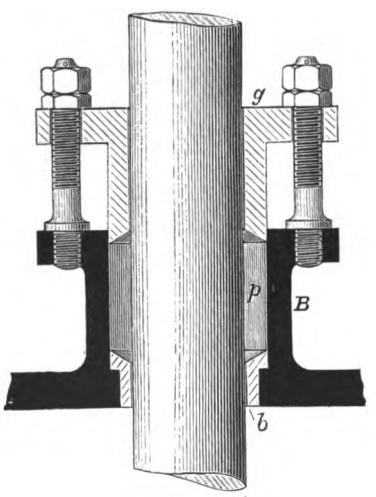


FIG. 45.

Into the space, *p*, thus formed around the rod, is placed some kind of packing until the box is nearly full. This is in the form

of a number of rings, each consisting of a length of packing just sufficient to encircle the rod. When in place, it is pressed into the box by screwing down the adjustable cover or gland, *g*; and being thus spread out laterally the packing grips the rod with increasing tightness as the gland is screwed farther in. The gland is of gun-metal, or is lined with gun-metal.

Packing.—The material used depends upon the circumstances. For moderate air or water pressure, hemp or india-rubber-core packing is commonly employed. The latter, consisting of a core of india-rubber wrapped in canvas, was also largely used with low-pressure steam; but such packing is quickly destroyed by high temperature and abrasion. Packing made of asbestos fibre has been much used for steam stuffing-boxes with high-pressure steam, but not with entire success in situations where it is exposed to heavy and continual friction, as in the case of the piston-rod and slide-rod boxes.

In these cylinder stuffing-boxes it is especially important that the packing used shall be one capable of working for a long time without requiring renewal. *Metallic packing* best answers the requirements, and is generally used for these boxes in modern engines. It is almost a necessity for the high-pressure cylinder, and is with great advantage fitted also to the other cylinders. This packing not only wears much longer, but works with less friction, and is less liable to score the rod than asbestos or any other kind of packing. Metallic packing is generally in the form of white metal rings of triangular section (W, Fig. 46), each ring made in two lengths; alternately with these are fitted gun-metal rings, G, of similar section; and, as the gland is screwed in, the white-metal rings are pressed on to the surface of the rod by the action of the inclined surfaces.

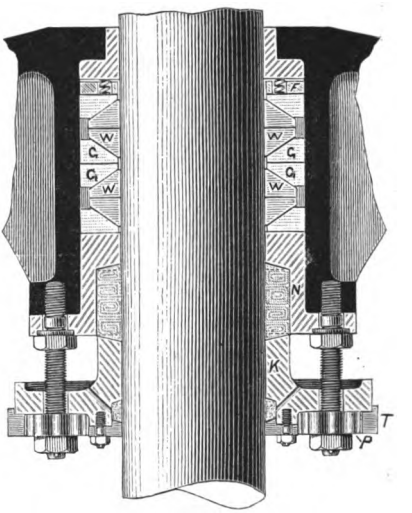


FIG. 46.

As generally fitted in the service, the gland, N, is screwed hard down on the stuffing-box, meanwhile compressing a number of spiral springs carried in a frame, *f*, at the bottom of the box. The pressure upon the rod is kept up during slight wear by the elasticity of the compressed springs. The gland is recessed to form a second box around the rod, for the reception of ordinary packing; this is compressed by a second or outer gland, K, the object being to check any leakage past the metallic rings.

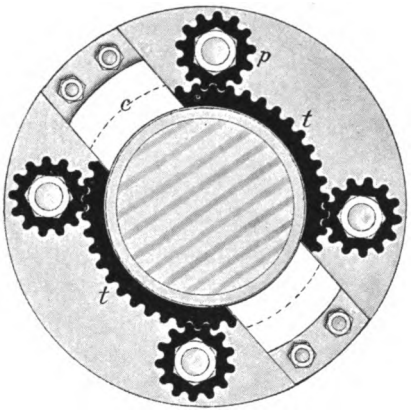


FIG. 47.

Adjusting Gear for Glands.—Glands are held in place by studs and nuts; and, in important glands, the nuts are so connected as to be screwed up simultaneously. The plan is to use toothed pinions (*p*, Fig. 47) as nuts, and to connect them together by a toothed ring, *t*, which is held up to the face of the gland by clips, *c*, and gears with the pinions. On screwing up one nut the others are screwed up also by the rotation of the ring; this is very convenient, and the gland is set up fairly. In the cylinder stuffing-boxes with metallic packing this plan is used for the outer gland.

Framing.—The engine framing is a fixed and rigid framework which holds the working parts in their proper positions relatively to each other. The cylinders and shafting in particular require to be so held that no *relative* movement is possible. The framing carries the necessary faces and bearings at the places where these and other parts are to be secured, and provides also the necessary guides, as well as the means of securing the engines in the ship. The details are subject to considerable variation, even in the same type of engine, and in different types they vary greatly. Attention will here be confined to the vertical type.

In vertical engines the principal framing consists of upright frames which support the cylinders, and of a horizontal bottom-frame or *bed-plate* to which the upright frames are bolted. The

bottom-frame consists of a number of athwartship girders similar to M, Figs. 33 and 54, carrying the main-shaft bearings, and of a number of fore-and-aft girders tying them together at their ends so as to form one frame. This bottom-frame is bolted down on the *engine bearers*, which are built up on the ship's frames, and are of a suitable form for receiving and supporting the several girders of which the bottom-frame is composed.

The upright framing, as usually fitted in service engines, may be seen in Fig. 33. It consists of large hollow castings, F, of iron or steel, at the back of the engines, supplemented by forged or cast steel columns in front,—an arrangement which leaves the working parts visible and accessible from the front. The columns, C, in the figure are of solid forged steel. The back frames are tied together at the upper part by substantial distance-pieces, to which they are bolted (see frontispiece).

On the top of each frame is a large face for the attachment of the cylinder (see plan of frame, Fig. 48); on the front is the guide, *g*, for the piston-rod head; and at the lower part the frame

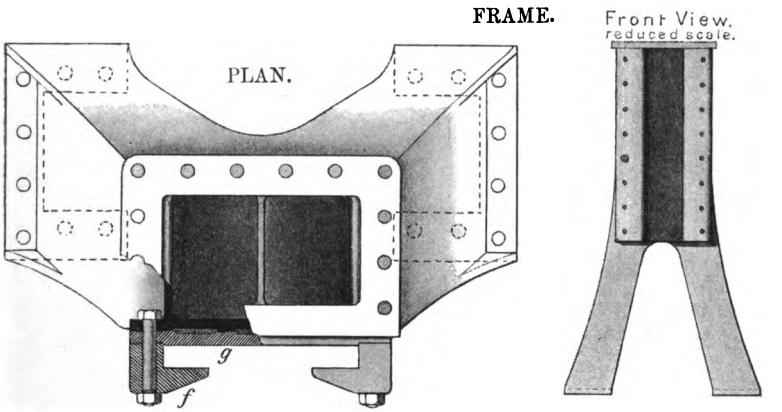


FIG. 48.

divides to form two feet (see front view and also frontispiece) by which it stands on two athwartship or main-bearing frames, one on either side of the crank.

In some cases, instead of the above, four smaller frames are used for each cylinder; these are all alike, and are placed symmetrically, two at the front facing, two at the back. The front is

not so open with this arrangement, though it is in other respects a good one.

Piston-Rod Head, and Connecting-Rod Bearings.—Coming to the gear outside the cylinder, and keeping to the direct-acting type, we find the head of the piston-rod formed suitably for the double purpose of connection with one end of the connecting-rod and of being guided to move in a straight line, and the connecting-rod formed suitably for connection with the piston-rod head and with the crank-pin by means of free-working joints or bearings at the ends.

The bearing in each case is formed by a pin and a pair of brasses enclosing it, the pin and brasses turning relatively to each other. The brasses are carefully fitted to the pin, and are capable of adjustment. The bearing at the inner or tail end of the connecting-rod is the *gudgeon bearing;* that at the crank end or head is the *crank-end bearing.*

Gudgeon Bearing.—The brasses for the gudgeon bearing are generally carried in the piston-rod head. The connecting-rod tail

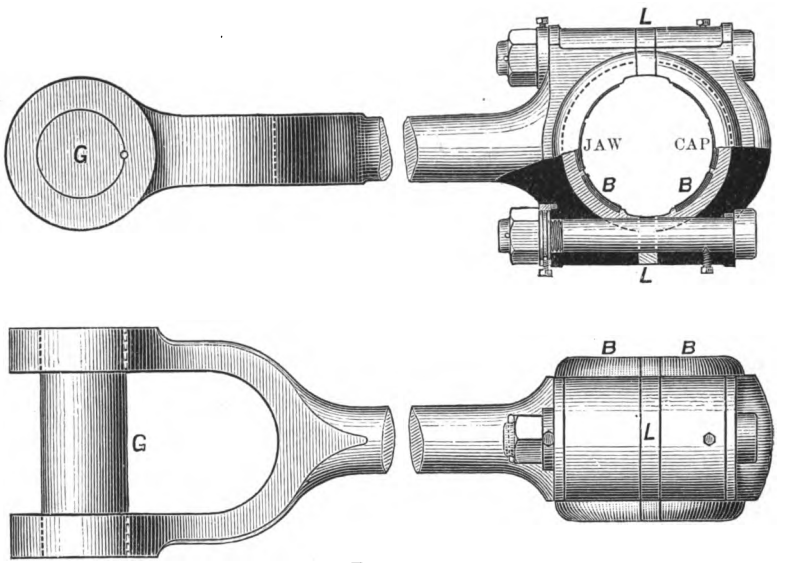


FIG. 49.

end is forked, as in Fig. 49, and carries the pin or gudgeon, G, which is made separately, and is shrunk or otherwise fixed in place.

The piston-rod head, Fig. 50, is forged in one piece with the rod, and is formed with a recess into which are fitted the gun-

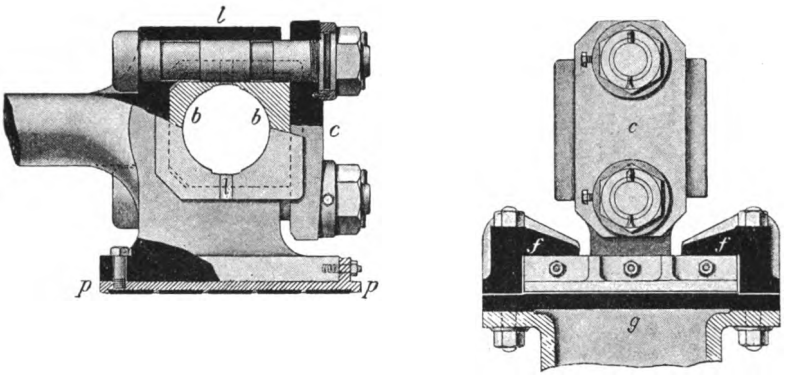


FIG. 50.

metal brasses *b*, *b*. The inner brass is cylindrical on the back surface, or flat as in the figure; the outer brass has a flat back against which the cap, *c*, bears; the bolts and nuts enable the brasses to be closed together over the pin; and the liners or fitting strips, *l*, between the brasses are distance pieces to admit of re-adjustment after wear.

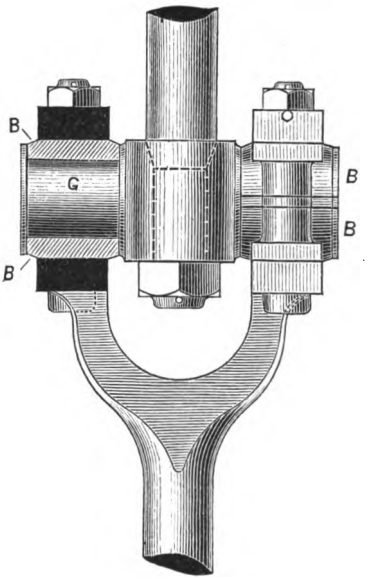


FIG. 51.

The above construction has been usual in direct-acting engines in the service, though an alternative plan, shown in Fig. 51, is now frequently used. The head is here a separate forging which carries the gudgeon in two parts, one on each side, and is secured to the piston-rod as shown. The connecting-rod end is forked, but is without a pin, carrying instead a pair of adjustable brasses, B, B, in each arm of the fork for attachment to the pin. In the figure the left-hand part of the bearing is in section, showing the half-length of gudgeon, marked G.

This construction may be seen in the set of engines shown in the frontispiece, and the more ordinary construction in Fig. 33.

Crank-End Bearing.—The brasses are here carried in the head of the connecting-rod, as in Fig. 49, and the pin is the crank-pin. The connecting-rod head is in two parts: the *jaw*, solid with the rod; and the *cap*, which is secured to it. A cylindrical aperture is bored through the head, half in the jaw of the rod, half in the cap; into this aperture are fitted the brasses B, B. These are gun-metal bushes, cylindrical on both outer and inner surfaces. Liners, L, are fitted for adjustment, and the whole is secured by bolts and nuts.

The brasses are lined on their inner surfaces with white metal, shown in the figure by the small dark patches. The velocity of rubbing is here considerable, and it is specially important to reduce the friction as much as possible. White metal is an alloy which works with very little friction, and is yet firm enough at ordinary temperatures to withstand the pressure at the bearing, if properly supported by gun-metal. It is liable, however, to soften and perhaps run, if from any cause the bearing should become heated. A spare set of brasses is carried for each rod, and the brasses so injured may be refitted with white metal on board.

The brasses of the gudgeon bearing are not fitted with white metal. The motion is here merely an oscillation, and with white metal both gudgeon and brasses rapidly wear oval. The gudgeon, however, requires to be case-hardened to work well with gun-metal.

Piston-Rod Guide.—The piston-rod head, Fig. 50, has a flat base on which is carried the slipper, *p*, a gun-metal plate faced with white metal, which is intended to slide on the surface of the cast iron guide-plate, *g*, Figs. 33, 48, and 50. This plate is secured to the back frame of the engine; and during the ahead motion the cross-thrust on the piston-rod head keeps the slipper pressed against it. The flanges, *f* (usually separate, and secured by bolts), resist the thrust in the opposite direction when going astern.

The ahead guiding-surface is scored with channels for the distribution of oil, and behind the guide-plate is usually a space, seen in Figs. 33 and 48, through which water is circulated to keep the plate cool.

When the alternative construction of piston-rod head, Fig. 51, is used, the guide is sometimes differently formed. The two parts

of the gudgeon are prolonged, and each part carries a *slipper-block* at the end. These blocks slide between guiding surfaces formed on the frames, which in this case are arranged in pairs facing each other in the manner referred to in the description of the framing.

Shafting and Bearings

General Description.—Fig. 52 gives an elementary view of the arrangement of the shafting in a twin-screw ship. The line of shafting consists of the *main-shaft* or *crank-shaft*, *abc*, of the engine, and of the *screw-shaft*, *ddd*, which runs aft from the main-shaft and carries the screw. In general both are made in lengths coupled together, the main-shaft in two or in three lengths, and the screw-shaft usually in three or four; and the whole is held in a number of bearings—*main bearings* for the crank-shaft, and *screw-shaft bearings* for the screw-shaft. The latter include the *stern-tube bearing* fitted where the shaft passes through the hull, and in twin-screw ships the bearing in the outer bracket, A, close to the screw, as well as inboard bearings, *n*. The main bearings, *m*, are commonly two to each crank, placed forward and aft of the crank as close to the arms as possible. The main-shaft, with its bearings, may be well seen in the frontispiece.

The shafting is of forged mild steel, and is hollow from end to end, the internal diameter being about half the external. The object in using hollow shafting is to obtain the necessary strength with less material than would be necessary with a solid shaft. The central portion of a solid shaft is the least effective in resisting twisting, and its removal may be compensated by a very slight addition to the diameter; for instance, a 16″ shaft with an 8″ hole is equivalent to a solid shaft only $\frac{3}{8}$″ less in diameter. The crank-pins are also hollow; and the saving in weight, in comparison with the equivalent solid shaft, is about 18 to 20 per cent.

Crank.—The crank used in marine engines is a *double crank*, consisting of two *crank arms*, CC, Fig. 53, and a *crank-pin*, P, the latter a trifle larger in diameter than the shaft. The whole is in one piece with the length of shafting, which is forged with a lump or slab of sufficient size at this part, and is afterwards machined and turned to the finished shape and size. The cores of the shaft and pins are removed by boring.

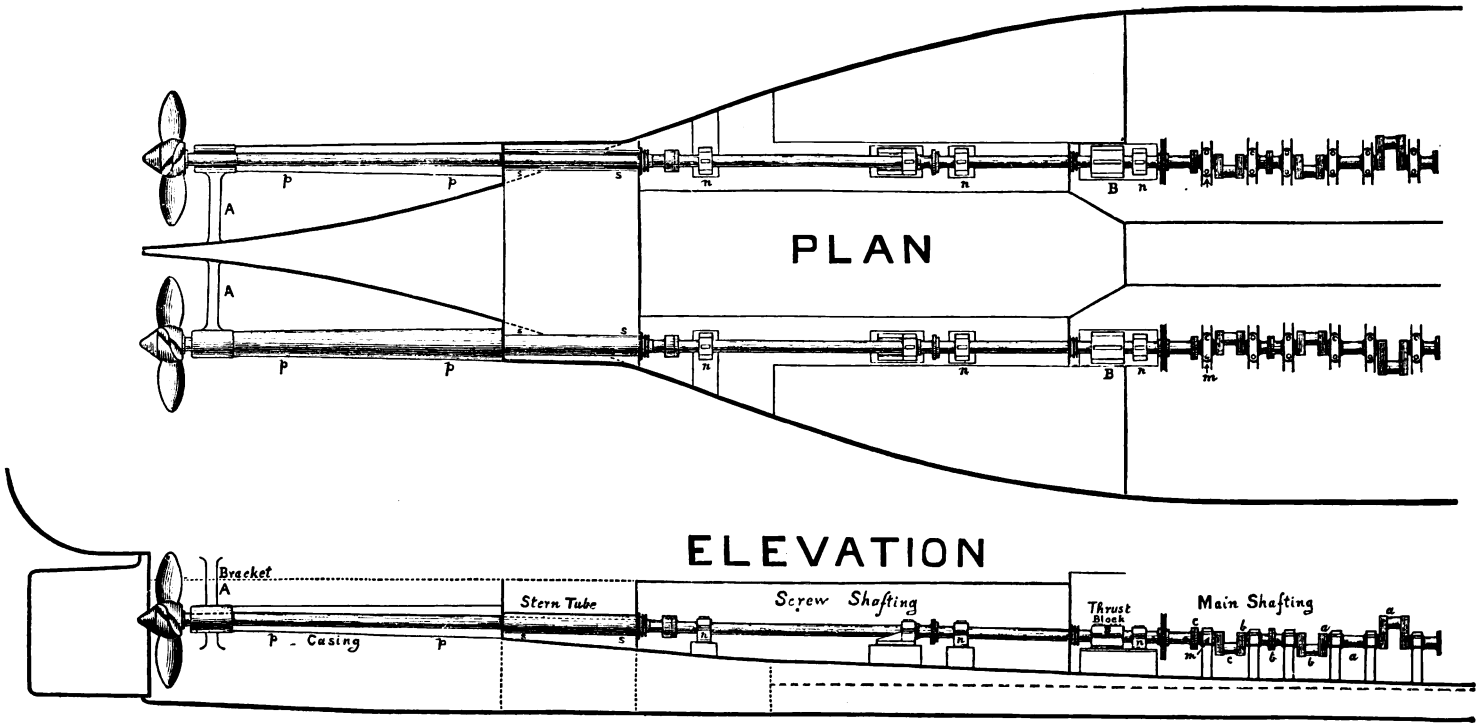

PLAN
ELEVATION
Bracket
A
P
Casing
Stern Tube
Screw Shafting
Thrust Block
Main Shafting
B
n
m
a
b
c

FIG. 52.

Shaft Coupling.—Adjacent lengths are coupled together by an ordinary flanged coupling, Fig. 53. Flanges, F, F', are forged solid with the shafting at the ends of each length; these are con-

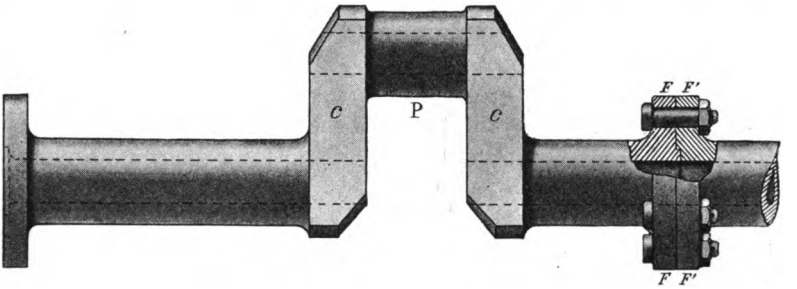


FIG. 53.

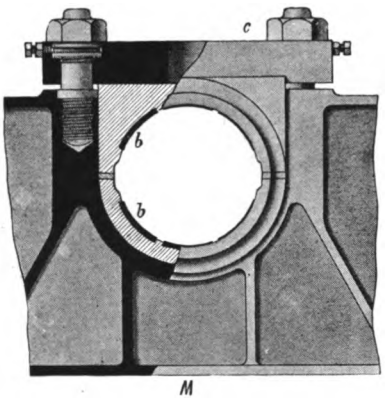
nected by large bolts, which are a driving fit in the holes and are secured by nuts. One length is filleted into the other as shown.

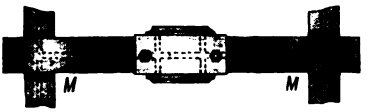
For simplicity the crank-shaft is made of the same diameter from end to end; but it may be noted that the after part is subjected to much more severe stress than the forward part, since the parts marked *c*, in Fig. 52, have to transmit the power of the three engines, while the corresponding parts *b* and *a* transmit only the power of two engines and of one engine respectively. Where practicable with three lengths, the after and one of the forward lengths are made interchangeable, so that a defective length aft may be shifted forward; and preferably, all three lengths are interchangeable.

The screw-shaft is free from the heavy bending forces brought by the steam-pressure upon the crank-shaft; the inboard part of it is therefore made of a reduced diameter; but the remainder, generally distinguished as the *stern-shaft*, running within the stern-tube and outside the ship, is as large as the crank-shaft; and where the outside part is of great length, as in cruisers, that part is larger still. The object is to secure additional immunity from the danger of rupture within the stern-tube bearing, and extra stiffness outside to compensate for the distance between supports.

Main Bearing.—Fig. 54 shows one of the main bearings of a vertical engine. The shaft-journal revolves within a pair of brasses, *b*, *b*, fitted into one of the transverse frames, M, forming part of the bed-plate. The frame is cast with a recess for receiving the brasses and shaft. The principal forces act

vertically; the brasses are therefore placed above and below the shaft. The bed for the bottom brass and the back surface of the brass itself are of cylindrical form, to enable the brass to be removed with the shaft in place; this is done by supporting the shaft and moving the brass around it to the upper position. The





Plan of Main Bearing Frame.
Reduced Scale.

FIG. 54.

outer brass has a flat back on which bears a simple flat cap, *c*, secured by bolts. The adjustment of the bearing is in a vertical direction, and is facilitated, as before, by adjusting strips between the brasses. The latter are lined with white metal. A hand-hole in the cap enables the brasses to be felt.

Bearings.—It will be seen that the object of the construction in this bearing and in the other important working bearings which have been described is that the shaft-journal or pin shall be enclosed by parts which are easily renewable as well as adjustable, and which are of a suitable material, possessing strength and firmness without brittleness, working with little friction, and little liable to injure the journal or pin. Gun-metal, lined on the working surface with white metal, answers these requirements well, except in the case of the gudgeon bearing, where, as has been stated, gun-metal alone is better, provided the gudgeon be case-hardened.

The white metal in the main-bearing brasses, crank-end brasses, and slipper of the piston-rod head, is usually held in dovetailed recesses in the face of the brass, as shown in the figures, the metal having been run in molten. In the finished state it stands just clear above the surface of the brass.

For good working the bearing surfaces require to be large, accurately fitted, well adjusted, kept perfectly clean, and continuously lubricated. For main bearings ordinary siphon lubricators are commonly used, in which the oil flows continuously from an oil-box, along cotton wicks or worsteds, by a slow siphoning process, and drops to the journal through oil-holes in the top

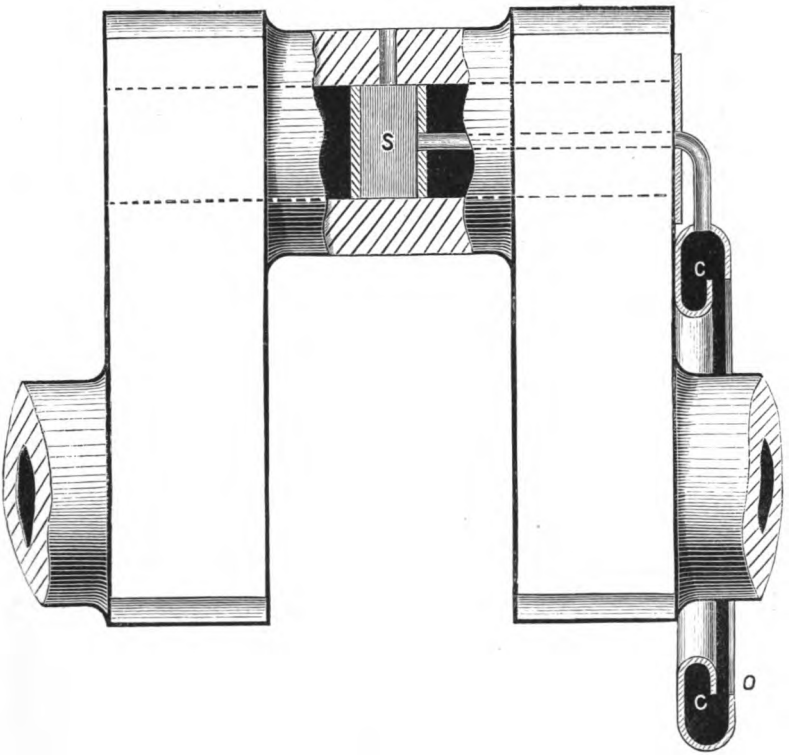


FIG. 55.

brass, reaching the various parts of the bearing by channels cut in the face of the brass. The piston-rod guide and the connecting-rod bearings are also supplied from lubricating boxes in various ways; but in the case of the crank-end bearing an additional lubricator of special form is provided, known as the *centrifugal* lubricator. This is shown in an elementary way in Fig. 55. It consists of an annular casing of the section shown, in two parts, secured together around the shaft and fixed to the crank-arm;

the casing, CC, communicates with the space S, in the hole through the crank-pin, from which small holes pass radially to the outside of the pin. During the working of the engines, oil supplied to the casing at the open side, *o*, passes by centrifugal action to the rubbing surfaces.

Most of the parts requiring lubrication are now supplied by lubricating pipes led down from a common oil-box fixed on the cylinder; a regulating screw is fitted on the box for each pipe, and the pipes receive the oil by open mouths, so that the supply to each part may be seen and adjusted. This is convenient, but extra care is required to prevent dirt gaining access to the bearings.

Screw-Shaft Bearings.—The ordinary screw-shaft bearing, having merely to support the weight of the shafting, is of a very simple description: a cast steel block without brasses, the lower part lined with white metal, the cap covering, but standing just clear of the shaft-journal.

Bearings in Stern-Tube and Outer Bracket.—These are the very important bearings provided where the screw-shaft passes through the hull, and again, in a twin-screw ship, just forward of the screw (see Fig. 52). Between them these bearings must support the weight of a great length of shafting and the weight of the screw. They are of special construction, suited to their situation under water and their inaccessibility. Any considerable wearing-down at these parts would cause severe straining of the shafting and involve the docking of the ship for repair. The peculiar feature of these bearings consists in the use of wood as the supporting material and of sea water as the lubricant.

The *stern-tube bearing*, the detail of which is shown in Fig. 56, is in two parts, situated at the forward and after ends of the *stern-tube*, a stout gun-metal tube SS, which is fixed in the hull, and through which the stern-shaft passes. In the Navy this stern-tube is enclosed and supported by the *shipbuilder's stern-tube*, TT, a tube of steel plate worked into the ship's frames during the construction of the ship; this tube carries bushes, *PP*, at the ends, which are accurately bored in place for receiving the stern-tube. The latter is thus conveniently supported on the frames of the ship; it is secured in place usually by a flange inboard, as in the figure.

In twin-screw ships the frames and plating of the ship are bossed out at the parts where the shafts pass, so as to carry the

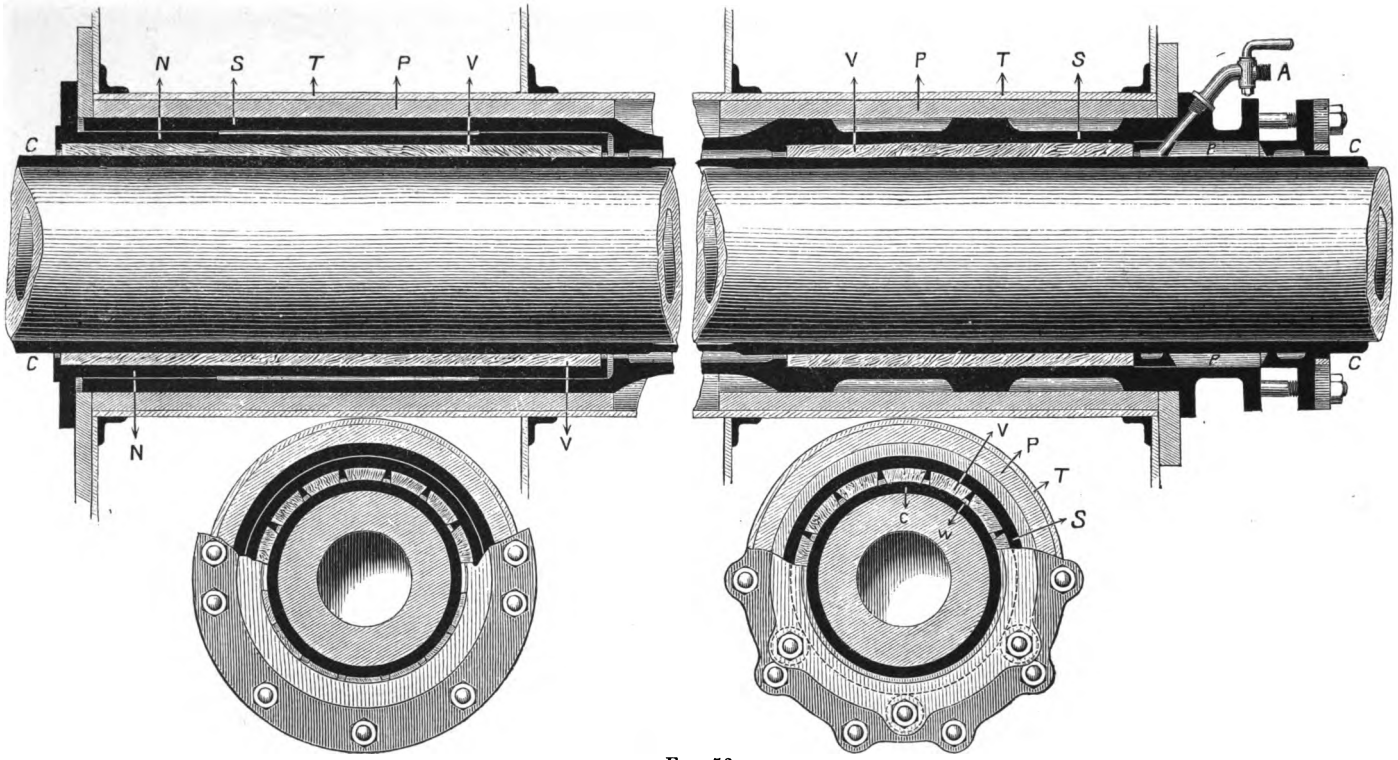


FIG. 56.

stern-tubes as far aft as possible. This is seen in Fig. 57, which shows the form at two sections; A at the mid-length, B at the after-end of the stern-tube.

The part of the stern-shaft which revolves within the stern-tube is cased in gun-metal, partly to protect it from the sea water which is admitted to the bearing, partly to provide a good bearing material, which may be renewed when worn. This casing or *sleeve* (CC, Fig. 56) is a tube long enough to cover the shaft from inboard to the after end of the stern-tube. It is usually shrunk on to the shaft, gripping it at each end for some distance, and is further secured to the shaft by a number of screwed rivets.

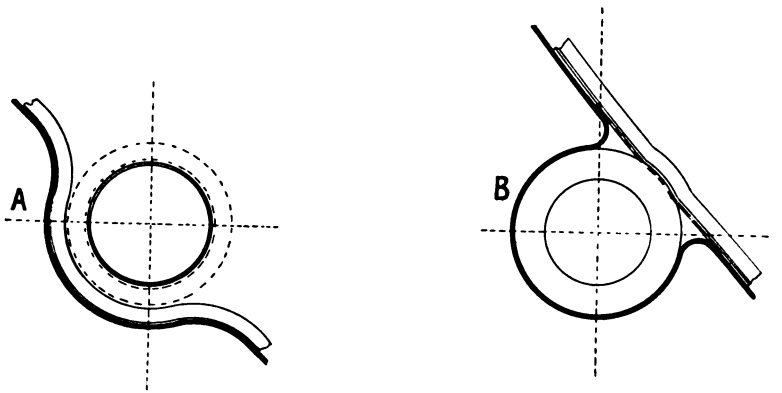


FIG. 57.

The shaft, so cased, runs in contact with a bearing surface of hard wood provided by inserting strips of lignum vitæ, V, V, into the stern-tube at each end. These strips, about 3″ wide, are held in dovetailed grooves cut longitudinally in the interior of the tube; and their inner surface, after being bored to the correct diameter, stands clear of the tube by about $\frac{1}{4}$″. Spaces, W (seen in the cross sections shown below the longitudinal section), are left between adjacent strips, by which sea water has free access to the bearing, filling the space between the shaft-casing and the stern-tube, and lubricating the wood at both ends of the bearing. On this thorough water lubrication the success of the bearing depends. Under ordinary circumstances the wear is exceedingly slight, but in shallow water sand passing into the bearing with the water is liable to cause rapid wear.

At the inboard end a deep stuffing-box is fitted to prevent a flow of water into the ship. Water may, however, be drawn from the water space for testing temperature by means of the cock A; and a slight flow may be allowed if necessary for keeping the bearing cool by slacking back the gland.

An improvement consists in fitting the wood strips at the after end into a separate bush, N, fitted into the stern-tube as shown in the figure. The bush is made in halves, and can be easily removed and refitted with wood if it should become necessary.

The bearing in the outer bracket (A, Fig. 52) is of the same description. A gun-metal bush is fitted into the boss of the bracket and secured by a flange; the bush carries lignum vitæ strips fitted in the manner already described, and the shaft is cased in gun-metal at the bearing.

Between this *after bearing* and the after end of the stern-tube, the shaft is not cased, except in sheathed ships, but is merely painted. A light casing of steel plate, *p*, is fitted to enclose the shaft and prevent the fouling of ropes. Water has free access to the interior of the casing. In sheathed ships the gun-metal sleeve extends from the forward end of the stern-tube to the screw, in order to protect the entire stern-shaft from galvanic action; and the light outer casing, *p*, is then also of gun-metal.

Thrust Bearing.—The bearings which support the shafting have now been described; there remains the thrust bearing, the office of which is to receive and to transmit to the ship the thrust produced along the line of shafting by the revolution of the screw. This bearing is situated near the forward end of the screw-shaft. In its simplest form, as in Fig. 52, it consists of a cast iron block with a removable cap, similar externally to an ordinary shaft bearing, but internally quite different, as will be seen by reference to Fig. 58, which is an elementary view of a portion of the block seen in plan with the cap removed and the shaft in place.

The shaft at this part is constructed with a number of collars, as shown, and the block has corresponding internal collars, which are alternate with the shaft collars. The latter press upon the after or forward faces of the internal collars, according as the screw is revolved for ahead or astern. In the one case the screw tends to force the shaft forward relatively to the ship,

in the other case to draw it aft; in either case the thrust is taken by the block, which is secured to a large and specially strong bearer built up on three or more of the ship's frames, and well secured in a fore and aft direction. A sole-plate, *p*, is bolted to the top of the bearer, and on this the block is bolted down, but is still adjustable in a fore and aft direction. The adjustment is made by large set-screws or wedges, *s*, *s*, at the ends of the block; and the bolt-holes, *h*, in the base of the block, are elongated to admit of the adjustment.

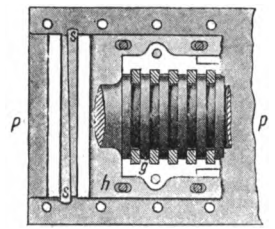


FIG. 58.

In this form of block the bearing surface is either of white metal, provided by covering the interior of the casting with that material, or in the older blocks, of gun-metal, provided by fitting separate gun-metal rings, *g*, into the casting. These rings are in two parts, bedded respectively in the cap and lower part of the block: they are easily renewed when necessary.

The thrust-block shown in Fig. 59 is now much more generally fitted than the simple form just described, its special advantage being that each collar may be adjusted independently of the others. The block, which consists of the lower part only without a cap, and which is a simple trough-shaped casting standing clear of the shaft collars, carries collars in the form of a number of horse-shoe shaped pieces, *h*, one of which is shown separately in front view; these are fitted over the shaft between the shaft collars, and are secured to two screwed bars, B, carried by the block, one at each side. Each collar is adjusted and secured in position on the bars by nuts, and is further secured and steadied by set-screws, S. The block may also be adjusted as a whole. The horse-shoe collars are lined with white metal on both faces. There is obviously no bearing surface for the bottom part of the shaft collars, but these are of larger diameter relatively to the shaft than in the block previously described.

Lubrication is provided separately for each collar of a thrust-block usually by means of ordinary oil-boxes; provision is made also for a copious supply of water in case its use should become necessary. Sometimes, in the kind of block just described, the trough formed by the casting is partly filled with oil, through

which the shaft collars run; the oil is thus carried up continuously to the bearing surfaces.

An ordinary shaft bearing, *n*, is fitted close to the thrust bearing,

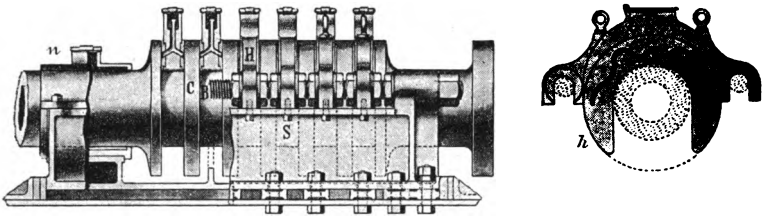


FIG. 59.

to support the shaft and prevent vibration and consequent unevenness of pressure on the collars.

CHAPTER V

SLIDE-VALVE

THE office of the slide-valve is to regulate the passage of the steam into and out of the cylinder. The requirements are as follows: Each stroke of the piston is to be produced by pressure exerted on one face, while exhaust takes place from the other; and both pressure and exhaust must change sides for a new stroke, in order that successive strokes may be produced in opposite directions. Some provision must also be made for preventing excessive shock to the connections at the end of each stroke.

The change from one stroke to the next necessitates four changes in the communications. At one end of the cylinder the communication must be closed to steam and opened to exhaust, and at the other end the communication must be closed to exhaust and opened to steam.

The simple slide-valve shown in Fig. 60 would produce these changes. The cylinder face shown has a single steam-port at each end, communicating with the cylinder by the steam-passages A and B; in the centre are the exhaust port and passage E. The valve has a flat face, which works in steam-tight sliding contact with the cylinder face. In the centre is a hollow or dished part C, the *valve cavity*, communicating with E; and at the ends are the *valve flanges* *a* and *b*. In the figure these are just long enough to close the ports, the valve being in the central position.

Steam is supplied to the space S at the back of the valve. If the valve be moved from its present position to the right, the port A will be opened to steam, and the port B to exhaust by way of C and E; if to the left, B will be opened to steam and A to exhaust.

The valve travels across the face alternately from left to right and from right to left; and in passing the central position the

valve shown would evidently reverse the communications. If, now, we imagine its movement to be so timed that it *passes the central position at the end of each stroke of the piston*, we have the changes produced at the end of the stroke. This would be managed by setting the eccentric, which gives motion to the valve, with its radius ⊥ to the crank, which moves with the piston.

The action described above would, however, be very defective. The changes necessary for the reversal of the stroke should not be delayed until the completion of the stroke, nor should they take place all together. With both pressure and exhaust continued

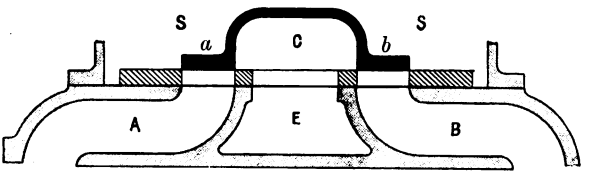


FIG. 60.

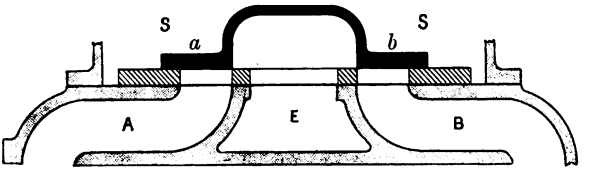


FIG. 61.

until the end of the stroke, there would clearly be nothing to arrest the piston except its mechanical connections with the crank-shaft, and these connections would suffer from the consequent jar and shock at the end of each stroke.

To avoid this, it is desirable, before the end of the stroke, to remove the pressure from the *driving side*, and to exert pressure on the *exhaust side*, so that the energy of the reciprocating parts may be absorbed by fluid pressure. This might be effected by setting forward the eccentric to make the valve pass the central position a little earlier; the changes would then occur just before the end of the stroke, but they would still occur together; and experience has shown that for efficient working they require to be separated. For instance, on the driving side there should be an interval between the closing of the port to cut off the supply of

steam and the reopening of the port to release the steam from the cylinder; and on the exhaust side there should be an interval between the closing of the port to cut off the exhaust and its re-opening to admit new steam for the next stroke. When so arranged, during the interval on the driving side the steam will expand and the driving pressure become reduced; and during the

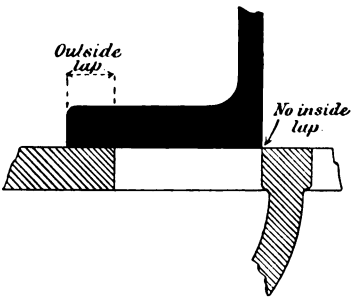


FIG. 62.

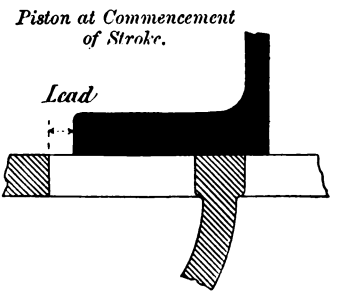


FIG. 63.

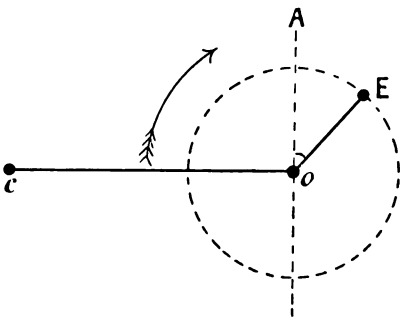


FIG. 64.

interval on the exhaust side the resisting vapour, which would have passed freely out of the cylinder in front of the piston if the port had remained open, being now imprisoned, will be compressed as the piston advances, and will thus offer an increasing fluid resistance to its motion.

Referring to Fig. 61, which shows a valve with *lap*, the valve being centrally placed, it will be seen that the valve-flanges are long enough to overlap the ports considerably on the outer edges. The effect of this is to produce an interval between the closing

and the reopening of each of the ports; and further, to make a considerable interval between the closing of one port to steam and the opening of the opposite port to steam.

Placing the valve in a position to the right with both ports open,—A to steam, B to exhaust,—and drawing it across the face to a similar position on the left, we see that the "cut-off" of the steam at A takes place first, the release of the driving steam at A and closing to the exhaust at B next, and the admission of new steam at B last; this is the proper order for the operations.

The last of these should take place just before the end of the stroke, so that there may be an opening to steam ready by the commencement of the return stroke. To secure this it is only necessary to set the eccentric with its radius sufficiently in advance of the crank.

Definitions.—Lap, outside or *inside,* is the extent to which the valve-flange overlaps the steam-port on the outer or inner edge of the port when the valve is in its mid-position: see Fig. 62. Inside lap is now generally either zero or a negative quantity: in the latter case the inner edge of the flange falls short of the edge of the port, instead of overlapping it.

Lead is the extent to which the steam-port is open to steam when the piston's stroke commences: see Fig. 63.

Angular advance of the eccentric is the number of degrees by which the angle between the eccentric radius and the crank exceeds a right angle: Fig. 64, angle AOE. This is now frequently about 40°.

Summary.—Action of Slide-Valve (see sketches and remarks, page 91).—At the commencement of each stroke the steam-port on driving side of piston is open slightly to steam, the opposite port is well open to exhaust, and the slide-valve is moving in the direction in which the piston is starting. During the early part of piston's stroke the valve moves in this direction far enough to uncover the port at one end fully to exhaust and the port at the other end about $\frac{2}{3}$ of its width to steam.

The valve then moves in the opposite direction, and, during the latter part of piston's stroke, closes, and afterwards reopens the ports at both ends, though not together; changing their communications from steam to exhaust and from exhaust to steam. Four operations are thus performed, as follows:—

Operation.		Usual Point of Stroke.	Effect.
Name.	How produced.		
"Cut-off"	By closing port on driving side to steam	About ·7	To give expansion and reduce driving pressure.
"Compression"	By closing port on exhaust side to exhaust	·85 to ·9	To absorb momentum of piston gently, by resistance of vapour cushion.
"Release"	By opening port on driving side to exhaust	·9 to ·85	To greatly reduce driving pressure, and prepare exhaust for return stroke.
"Admission"	By opening port on exhaust side to steam	Just before end of stroke	To complete action of compression and bring full pressure on piston in time for commencing return stroke.

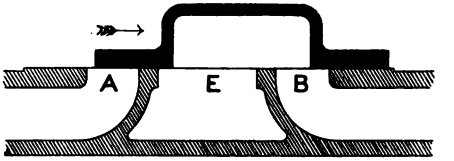
The point of the stroke usually named for any given operation is the mean between the point for the *outstroke* (stroke towards the shaft, or downstroke in vertical engines) and for the *instroke*.

The obliquity of the connecting-rod to the line of stroke causes the piston to be more advanced in the one stroke, and less advanced in the other, than it should be to correspond exactly with the crank's position; so that the operations, which practically depend on the crank's position, do not occur at the same points of the respective strokes, though the valve be symmetrically set. The obliquity varies from zero at the dead points to a maximum when the crank is midway between the dead points, that is, at about midstroke.

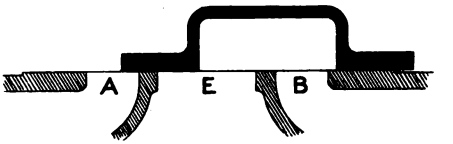
The effect, in direct-acting engines, is to cause the cut-off, and in a less degree the other operations, to occur at earlier points of the instroke than of the outstroke.

In horizontal engines, usually no notice is taken of this inequality; and the valve is so set on the rod that at its midstroke it stands symmetrically over the ports, its centre coinciding with the centre of the exhaust-port. In vertical engines, however, this would mean that steam would be admitted later, and would be cut off and exhausted earlier, on the under side of the piston than on the upper side, the compression also being less

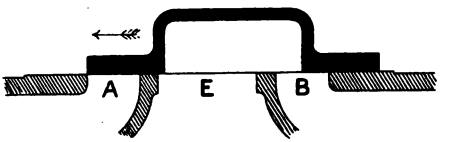
Commencement of piston's stroke from left to right. Valve moving in same direction.



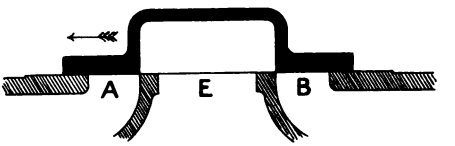
End of valve's travel to right. Valve commences to return.



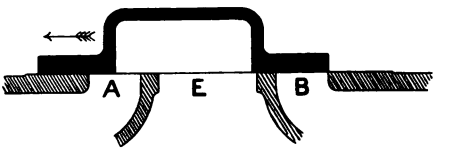
"Cut-off" at port A.



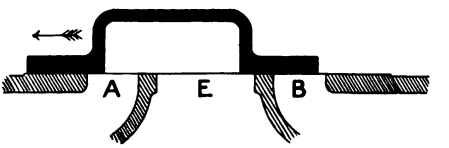
"Compression" at end B, "Relea se" at A.[1]



"Admission" at port B.



End of piston's stroke from left to right. Piston commences return stroke.



[1] "Compression" and "release" occur together only when the inside lap is nil, as is here assumed. With *positive* inside lap, "compression" occurs before "release"; with *negative* inside lap, "release" occurs before "compression."

under the piston than above it; while the exact reverse of this is what is desirable, on account of the weight of the moving parts, which here assists the steam in the downstroke and resists it in the upstroke. In vertical engines, therefore, the valve is set slightly higher on the rod, so that the centre of its travel is somewhat above the middle of the exhaust-port. If the effect of thus placing the valve above the symmetrical position on the face be considered, it will be seen that steam is admitted earlier, and is cut off and exhausted later, on the under side of the piston than before, and the reverse of this on the upper side; and at the same time compression is increased below the piston and diminished above it. Thus all that is required is obtained. The flanges of the valve have been assumed to be of equal lengths, so that the lap at the two ends would be equal with the valve placed centrally on the face; the actual lap is therefore unequal at the two ends in vertical engines, measured with the valve in mid-position, that is at midstroke.

Similar effects may be produced by making the upper and lower flanges of unequal length.

Improvements for Ease in Working.—The simple valve previously described requires a long stroke; and also, if no relief be afforded to it, works with great friction, on account of the steam-pressure in the valve-chest pressing it hard against the cylinder face. This involves a considerable expenditure of power and undue wear of the valve and cylinder faces, as well as a heavy load on the eccentrics and gear, and the use of very large eccentrics. The following improvements are therefore in general use:—

1. A *double-ported* or *treble-ported* slide-valve is used, which requires only one-half or one-third the stroke of the corresponding single-ported valve; and an *equilibrium ring* is fitted at the back of the valve, which largely reduces the area exposed to steam-pressure, and so reduces friction.

2. An alternative plan is to use a *piston slide-valve*: a valve with a cylindrical face instead of a flat face. The pressure is here balanced, and the valve works with very little friction; it also requires only a short stroke.

These plans, by reducing friction and shortening the stroke, reduce wear and tear, spare the eccentrics, and save power. A more detailed description of them follows. The majority of slide-

valves in marine engines are double-ported, flat-faced valves, fitted with equilibrium rings; but high-pressure cylinders are commonly fitted with the piston-valve, and occasionally intermediate cylinders also.

Double-ported Slide-Valve (Fig. 65).—The cylinder face is double-ported, *i.e.* the steam-passage at each end of the cylinder communicates with two steam-ports, *s*, *s*, in the face, instead of

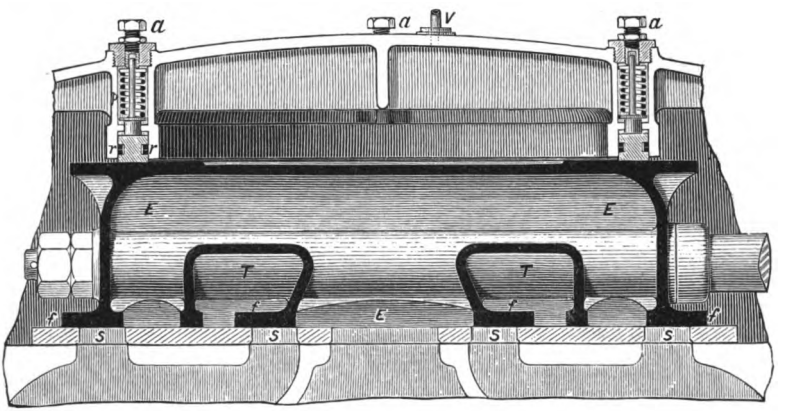


FIG. 65.

one port only, and the valve is correspondingly formed with two flanges, *f*, *f*, at each end, which act together and in precisely the same manner over the two ports.

The valve being centrally placed on the face, the four flanges stand over their respective ports with a certain amount of lap. The inner flanges form part of two steam-tubes, T, T, cast in the valve, running through it from side to side, communicating with the steam space and supplying the inner ports with steam. The space EE around the steam-tubes is in communication with the exhaust.

Placing the valve in position to the right, and drawing it from right to left, the same succession of operations will occur as before during the piston's stroke from left to right; the only difference being that two ports are now opened or closed simultaneously, so that one-half the previous movement is sufficient to produce any given opening for steam or for exhaust.

If, as is sometimes the case, three ports and three flanges are used at each end, the valve is a treble-ported valve and the

necessary travel is reduced to one-third. In this case there is a steam-tube for each of the four inner flanges.

Equilibrium or Relief Ring.—This is a ring fitted steam-tight between the back of the valve and the cover, and serving to exclude the steam from the greater part of the valve area. The ring is either carried on the back of the valve and pressed by springs against the inside of the cover, or is carried on the inside of the cover and pressed against the back of the valve. In the first case the ring moves with the valve, in the second it is stationary with the cover; in either case the surface of the ring

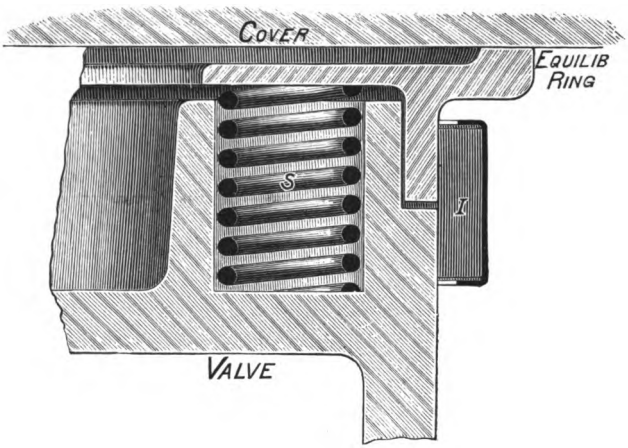


FIG. 66.

and the surface against which it presses are steam-tight rubbing surfaces. The steam in the valve-chest is thus confined to the space outside the ring, and the space within it is placed in communication with the exhaust. The valve is therefore exposed principally to the exhaust-pressure on both the inner and outer surfaces, though sufficient area exposed to steam-pressure must be left to keep the valve up to the face. The fittings vary considerably in detail. In Fig. 66 the ring, carried by the valve, is pressed against the cover by compressed springs, S. In Fig. 65 the ring, held in a groove in the cover, is pressed against the valve by springs, *p*, and adjusting screws, *a*; and in Fig. 67 is shown the detail for a similar arrangement previously indicated in the sketch of the cylinder. The ring is more generally

carried by the cover than by the valve, being in that case capable of adjustment from outside at any time.

Leakage of steam must be prevented at the part where the ring is held by the valve or by the cover, as the case may be; this is the object of the thick band of india-rubber, I, Fig. 66, and of the Ramsbottom rings, *r*, let into the sides of the equilibrium ring in Fig. 65. A good fitting for the purpose is that shown in Fig. 67, where a flexible copper ring, of the section shown at *c*, is secured by its flanges to the equilibrium ring and to the cover, the plan being similar to that used for the end of the cylinder liner.

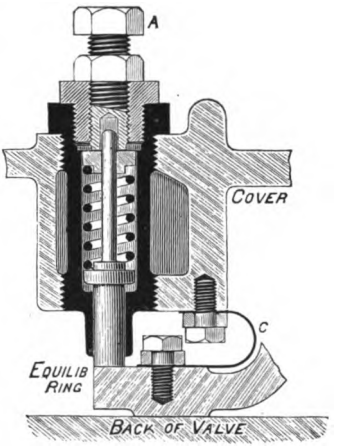


FIG. 67.

Where exposed to high pressure, as on high-pressure cylinders, equilibrium rings are liable to give trouble, and the flat-faced valve described is here generally replaced by the cylindrical or piston valve.

Piston Slide-Valve.—In Fig. 68 the cylinder face, CC, is cylindrical, instead of being flat as before; and the valve also, shown separately, is of circular section transversely, while the section from end to end is similar to that of an ordinary single-ported slide-valve. Referring to the sketches, Fig. 69, A represents a cylinder with an ordinary flat face, *ff*, across which are straight ports; while B represents a cylinder with a cylindrical face, *cc*, around which the ports run: *ss* is the boundary of the steam-passage at each end connecting the port in the face with the cylinder. The steam passes as shown by the arrows.

The face, *cc*, may be imagined to have been produced by curving the flat face, *ff*, until it becomes cylindrical, except that the actual face is larger than would thus be produced; and the cylindrical slide-valve may be similarly deduced from the flat-faced valve, the face of the valve being external, while the face of the cylinder is internal. The detail is shown in Fig. 68. The valve is hollow from end to end, and, usually, the ends and inside of the valve are in communication with the steam supply, while the space, EE, around the central part of the valve is in

communication with the exhaust, though the arrangement is sometimes the reverse.

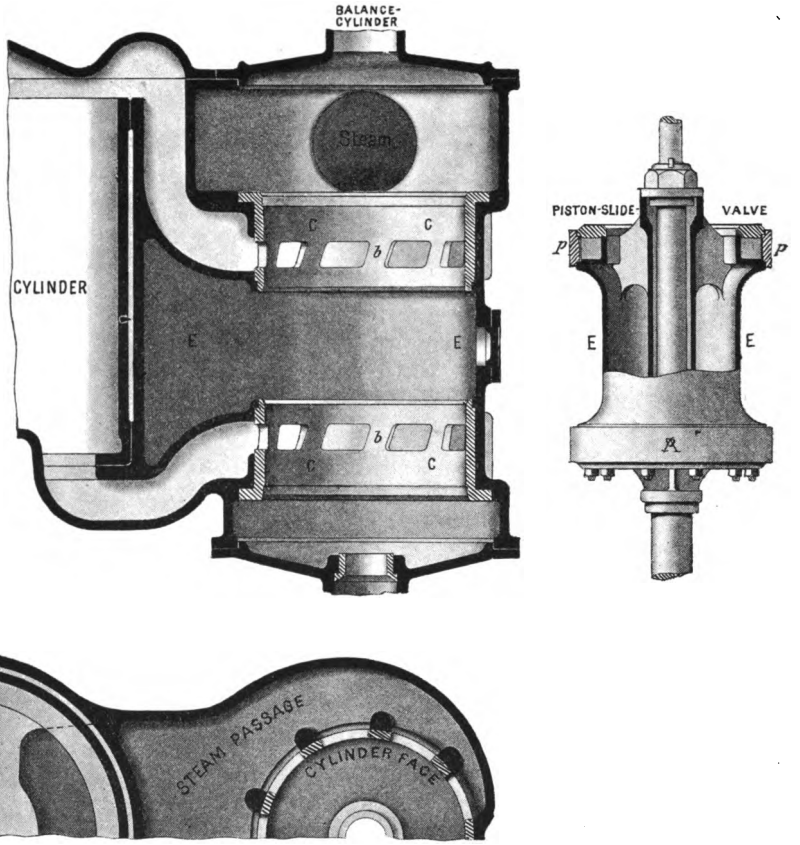


FIG. 68.

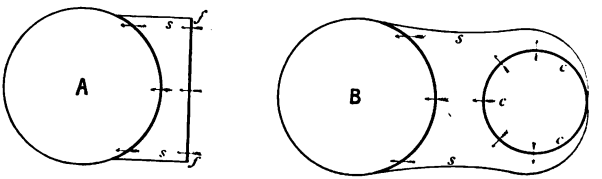


FIG. 69.

In the cylinder face is a single steam-port at each end. The flanges of the valve, which work over these ports, usually consist

of packing-rings, *p*, which exert a slight pressure on the cylinder face. The bars, *b*, across the ports are necessary for guiding and restraining the packing-rings when they are passing the ports.

This valve, being of circular section, is entirely balanced so far as steam-pressure is concerned, and the only friction is that arising from the necessary tightness of the flanges. The stroke required is short, because the ports are much longer than they would be if formed in a flat face on the same cylinder; they therefore do not need to be uncovered to the same extent.

Thus the piston-valve fulfils the requirements both of shortened stroke and of reduced friction. It is, however, difficult to keep steam-tight, and it occupies a considerably increased space; its use is therefore generally confined to high-pressure cylinders, where the difficulty in using the flat-faced valve is most felt.

In this connection may be mentioned the *balance cylinder and piston*, *b*, Fig. 35, fitted above the slide-valve in vertical engines. The object is to relieve the eccentrics of the extra load due to the weight of the slide-valve. This is done by giving the balance piston sufficient area, and allowing the steam in the valve-chest free access to the under side, while the upper side is kept in communication with the exhaust.

CHAPTER VI

SLIDE-VALVE GEAR—STARTING AND REVERSING ARRANGEMENTS

Eccentric and Rod.—Slide-valves are commonly worked by means of eccentrics rotating with the crank-shaft. The eccentric and rod is a piece of mechanism largely used in machinery for converting rotatory motion into *reciprocating* motion, by which is meant a to-and-fro straight-line motion, such as a slide-valve requires. It is used instead of the *crank and rod*, of which it is in fact merely a modification, though it has advantages which make it preferable where the load upon it is not too heavy and the stroke required is short.

The mechanism consists of the sheave or pulley, *p*, the strap, *s*, and the rod, *r*, Fig. 70. The eccentric sheave is a disc keyed to the shaft so as to revolve with it, but fitted non-centrally, the centre of the sheave being at some distance from the axis of the shaft. The strap, in two parts, embraces the circumference of the sheave, to which it is well fitted and adjusted so as to form a working bearing; the rod is secured to the strap, and at the other end is connected with the piece which requires reciprocation.

As the shaft revolves, carrying the sheave with it, the latter turns within the strap, which cannot revolve, but accompanies the sheave in its travel; and the common centre, E, of sheave and strap moves in a circle, shown dotted, about the axis, O, of the shaft. The outer end of the eccentric-rod is guided to move in a straight line pointing to the shaft centre, and it will obviously be moved forward and backward along a path equal in length to $2 \times OE$. The distance OE is called the *eccentric radius*, and represents the length of the virtual crank.

The sheave is in fact virtually a short crank-pin of sufficiently

large diameter to enclose the shaft; the rod and strap are a light connecting-rod and pair of brasses, and the action corresponds to that of a crank, OE, and rod used to produce reciprocation.

The detail of a marine engine eccentric may be seen by reference to the link-motion drawing, Fig. 75. The sheave is of cast iron or cast steel, and is in two parts secured together around the shaft, as shown; this is generally necessary to allow it to be placed in position. The line of division is of course on the shaft diameter, and is at right angles to the eccentric radius, so that the two parts are very unequal in size. The smaller part is frequently a forging, for strength. The strap is sometimes of gun-metal, though now more generally of steel (preferably

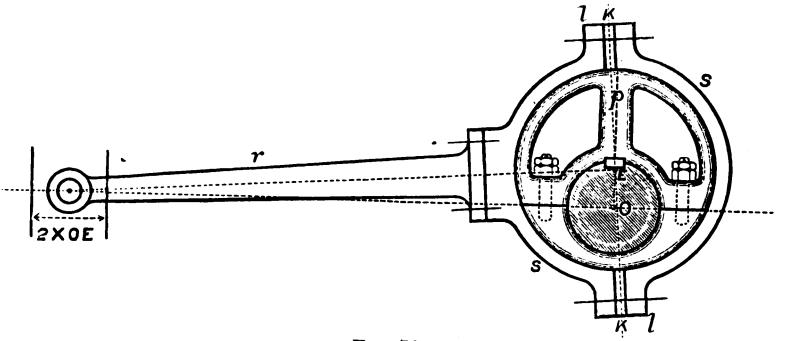


FIG. 70.

forged steel), lined with white metal. It is provided with flanges which prevent slipping sideways, the sheave being correspondingly formed, as shown in the cross section. The two parts of the strap are secured together around the sheave by bolts passing through lugs or flanges, *l, l.* Fitting strips, *k,* allow adjustment to be made. The eccentric-rod is of forged steel. It is secured by a flange at one end to the sheave; the other end is forked for connection with the link.

Link-Motion.—Marine engines are required to run both ahead and astern, and they should be so arranged that the change from one direction of rotation to the other may be made with ease and promptness. In practice the reversal of the engines is produced by a suitable change in the action of the slide-valves, and the slide-valve gear is specially adapted for this purpose.

The gear in general use is that known as the *link-motion.*

The proper position of the eccentric radius with reference to the crank has already been indicated in Fig. 64 for one direction of rotation. In Fig. 71 this position is represented by OE; and the similar position, OE′, on the other side of the crank, is that which an eccentric must occupy in order to cause the ports to be opened and closed correctly for rotation in the opposite direction. In either case the eccentric may be said to *lead the crank in the direction in which rotation is desired,* as shown by the arrows.

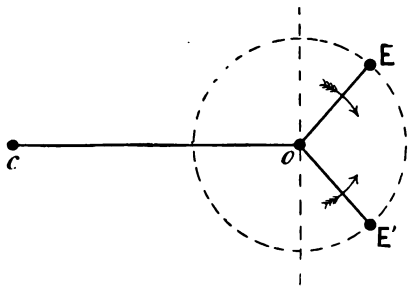


FIG. 71.

In the link-motion two eccentrics are used, occupying respectively the two positions above indicated: one the correct position for ahead working, the other the correct position for astern. The link and the gear connected with it afford a simple means of throwing either of these eccentrics into gear at will, and at the same time of throwing the other eccentric out of gear.

The eccentric-rod ends are attached to the ends of the link, and are therefore kept at a fixed distance apart. The link, *ll*, Fig. 72, is a bar with a curved slot to which is fitted a block, *b*; this block is connected with the slide-valve rod by a pin, *p*, about the axis of which it can turn freely. The link may be moved past the block, and held in any position, by the *suspension-rod, s*, which is actuated by the *reversing gear* to be afterwards described.

With the reversing gear set in any given position, the outer end of the suspension-rod is for the time a fixed centre, about which the rod swings during the working of the engine, the inner end holding and accompanying the link in its travel; and the block remains at nearly the same point in the link while the reversing gear is so set. By moving the reversing gear the link may be drawn into any desired position with reference to the block. If the link be placed in either extreme position, one of the eccentrics is brought into full gear, with full control of the valve's motion, while the other is put entirely out of gear. The eccentric in full gear is the one whose rod bears directly on the block (upper eccentric-rod, Fig. 72); and the valve receives

its motion from this eccentric as though the other did not exist. By simply moving the link to the other extreme position, the control of the valve is taken from the one eccentric and given to the other; with the result that as the link passes from one position to the other the engine is first stopped and then reversed. To effect this reversal is the primary object of the link-motion.

In the figure it will be noticed that the block, though in the "full gear" position, is not quite at the end of the slot. The reason for this is that while the link is attached to the end of the suspension-rod, which moves in the arc of a circle, the block

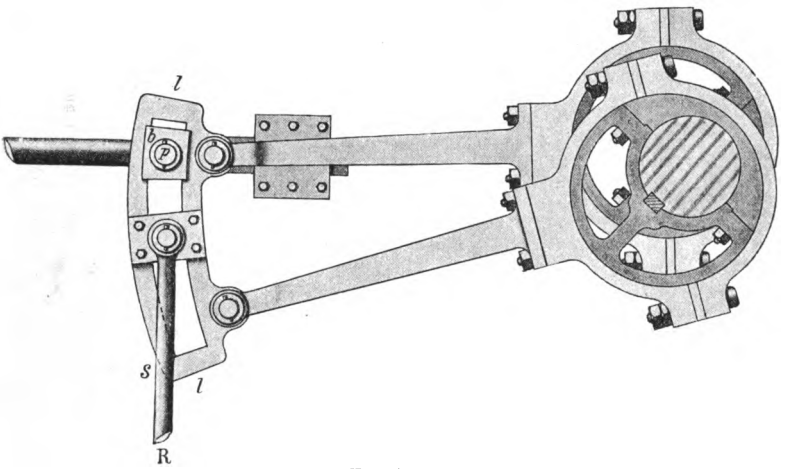


FIG. 72.

attached to the slide-valve rod moves in a straight line; there is therefore more or less movement of the link past the block during the working of the engines, and sufficient clearance to provide for this is necessary.

Linking-up.—In *full gear*, say for ahead, the valve will receive the full travel due to the ahead eccentric, and its operations will take place according to the setting of that eccentric and the proportions of the valve, as already explained.

Let the link be now moved so that the block occupies some position, A, Fig. 73, between the ahead end and the centre of the link. The ahead eccentric will still have the principal control of the valve, and the engine will still run ahead; but the astern eccentric will also have come to some extent into action, with the

result that the valve travel is now less, and the operations of the valve are earlier, than in full gear.[1]

The explanation of the valve operations being earlier is that the astern eccentric, OE', *when considered with reference to the ahead direction of rotation*, has very great angular advance in that direction; its effect, therefore, is practically to add to the actual angular advance of the ahead eccentric. Cut-off is the operation most affected by this increase in the angular advance. The other operations are advanced also, but to a smaller extent. With respect to the travel, it is obvious that the full stroke due to the eccentric will not be given in the direction of the valve's motion unless the eccentric is in full gear.

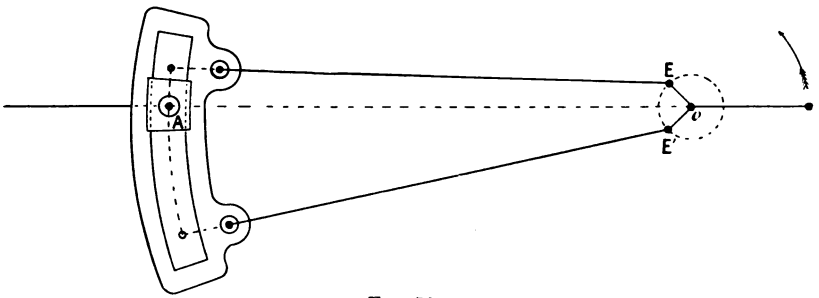


FIG. 73.

As the block is placed nearer the centre of the link the effects referred to become more marked, until, in the central position, the valve travel is largely reduced, and the operations are so much in advance of the piston's requirements that no stroke can be completed.

Centring the links has thus the effect of stopping the engines; while by "linking-up," that is, by making the block approach the centre, we are able to advance the point of cut-off, and consequently to increase the expansion given to the steam to a moderate

[1] It is assumed for simplicity that OE is the ahead eccentric. This would be the case in a return-connecting-rod engine, the direction of rotation being that shown by the arrow. In a direct-acting horizontal engine OE' would be the ahead eccentric; and the link would be up for ahead if connected with the eccentrics as shown in the figure. If, however, the eccentric OE were connected with the *lower* end of the link and OE' with the *upper* end, so that the eccentric-rods were crossed in the position of the gear shown, the link would then be down for ahead, as shown, in direct-acting engines. If so connected the link would be said to have "crossed rods"; while if connected as in the figure, *i.e. with the rods uncrossed when the eccentric radii point towards the link*, it has "open rods." Open rods are preferable for working "linked-up," the desired alteration in the cut-off being effected with less reduction of travel.

extent; and this is now the ordinary plan of working at reduced powers. The principal objection to linking-up beyond a moderate degree consists in the reduction of travel, and the consequent contraction of the port openings both for steam and exhaust.

Forms of Link.—Three forms of Stephenson's link may be found at work in the service: the *open* or *slotted* link, the *solid-bar* link, and the *double-bar* link. The last is the form now in general use.

In the slotted link, Fig. 72, the block is fitted into the curved slot passing down the centre of the link, and is free to turn about the pin, *p*, which connects it with the slide-valve rod. In the solid-bar form the link is a simple curved bar passing through the block, which is in the form of two sectors bearing upon its opposite edges, and cylindrical on their outer surface. This block is carried in a cylindrical recess in the slide-rod head, so that it can turn about its axis and give the necessary freedom to the link.

In the double-bar variety, Fig. 74, two curved steel bars, BB, are secured together, as shown, to form the link; distance-pieces, D, being fitted between the bars. The block consists of a steel

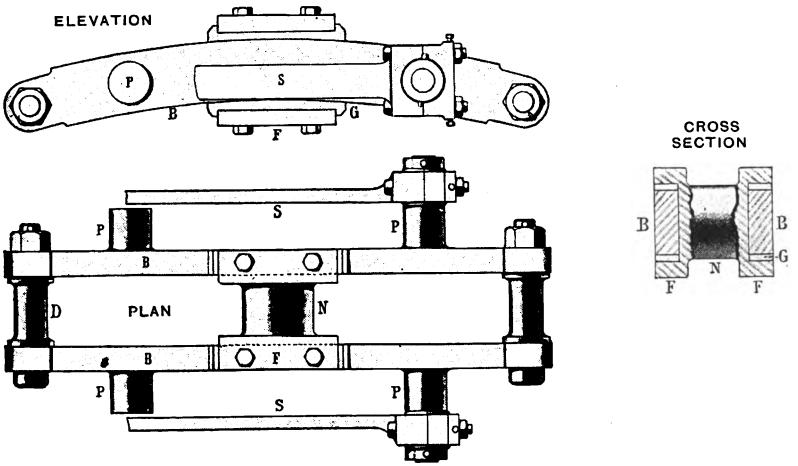


FIG. 74.

pin, N, situated between the bars, carrying flanges, F, which overlap and bear upon the edges of the bars, the flanges being faced with gun-metal strips, G, to provide a suitable bearing material.

The head of the slide-rod is fitted like a connecting-rod head with a pair of brasses for connection with the pin, N; and the latter,

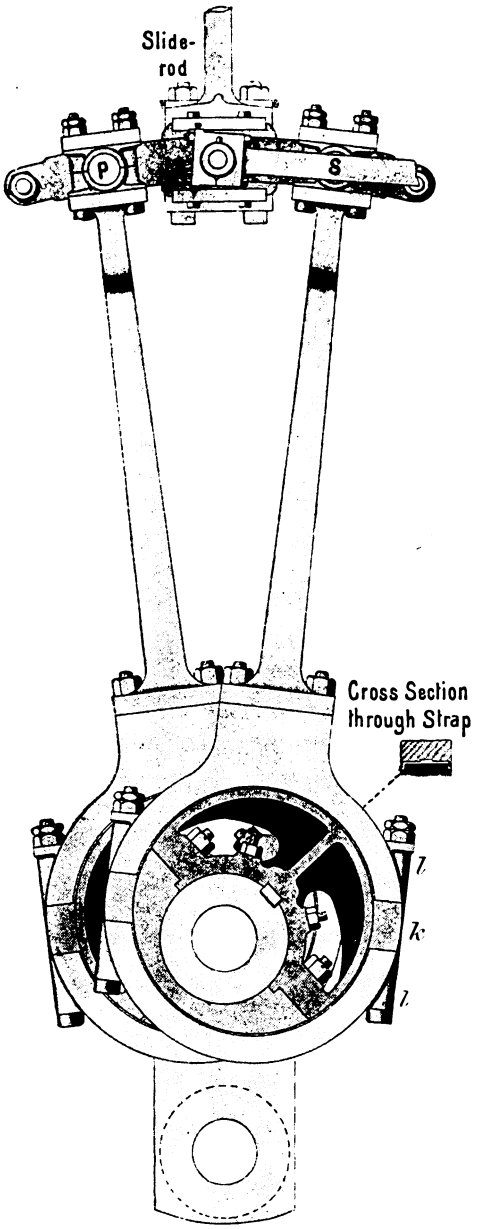


FIG. 75.

moving with the link, oscillates freely within the brasses, as may be necessary. This link is shown with the eccentrics and slide-rod connected in Fig. 75. The eccentric-rods have forked ends fitted with brasses, by which they are attached to the pins, P, P. The suspension-rods, S, generally two with this kind of link, are attached to the links at the middle or end, the pins in the latter case being prolongations of the pins, P, P.

The principle in the three forms of link described is identical. The curvature given should be such that the slide-valve will travel about the same mid-position in all positions of the link. This is approximately the case if the centre line of the link be a circular arc of radius equal to the length of the eccentric-rod, the latter being measured from centre line of link to centre of eccentric.

In the centre line sketch, Fig. 76, the length AC or BD being

given, and also the length or span of link (chord CD, usually $5\frac{1}{2}$ to 6 × OA, the actual proportions between these limits not being of material consequence), we get the points C and D. Taking C as a

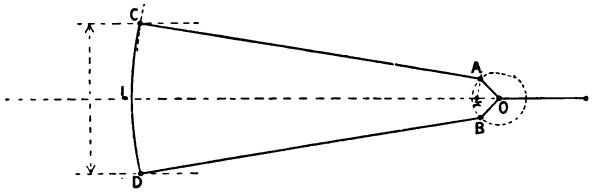


FIG. 76.

centre, and striking the centre line, OL, with radius CA, the point x so found is the centre from which the link curve, CLD, is drawn.

Reversing Gear.—This name is usually applied to the gear for moving the links, though in its fullest sense it includes also the link-motion already described. The arrangement in general use is shown by the elementary sketch, Fig. 77.

The gear for moving the links consists of (1) the *weigh-shaft*, W, supported in bearings, and carrying *reversing arms*, A, which correspond in number with the links, and are connected with them by the suspension-rods; (2) a worm and worm-wheel for turning the weigh-shaft through an angle sufficient to draw the links over into either extreme position; (3) the hand-wheel, H, and the steam starting-engine, E, either of which may be used for actuating the worm gearing. The starting-engine does the work of reversing much more quickly and conveniently than hand-power, but the hand-wheel is frequently of service.

Formerly the worm-wheel was *carried on the weigh-shaft*, and there was nothing in the arrangement of the gear to prevent the starting-engine from injuring it by running it too far; it therefore became necessary to provide the engine itself with automatic stopping gear, which should come into action when the links had gone far enough.

These arrangements, however, are liable to give trouble, and they have been superseded by the plan shown in Fig. 77, which entirely obviates the danger referred to.

This consists in placing the worm-wheel, M, in an independent position, and connecting it with the weigh-shaft by the rod, R, which couples together a pin, p, on the face of the wheel and the extremity of a reversing lever, L, keyed to the weigh-shaft. On rotating the worm-wheel the weigh-shaft will be turned in its

bearings; but the angle of swing of the reversing arms is obviously limited by the proportions of the pin-radius, *cp*, and the reversing lever. The extreme angle is just sufficient to carry the links

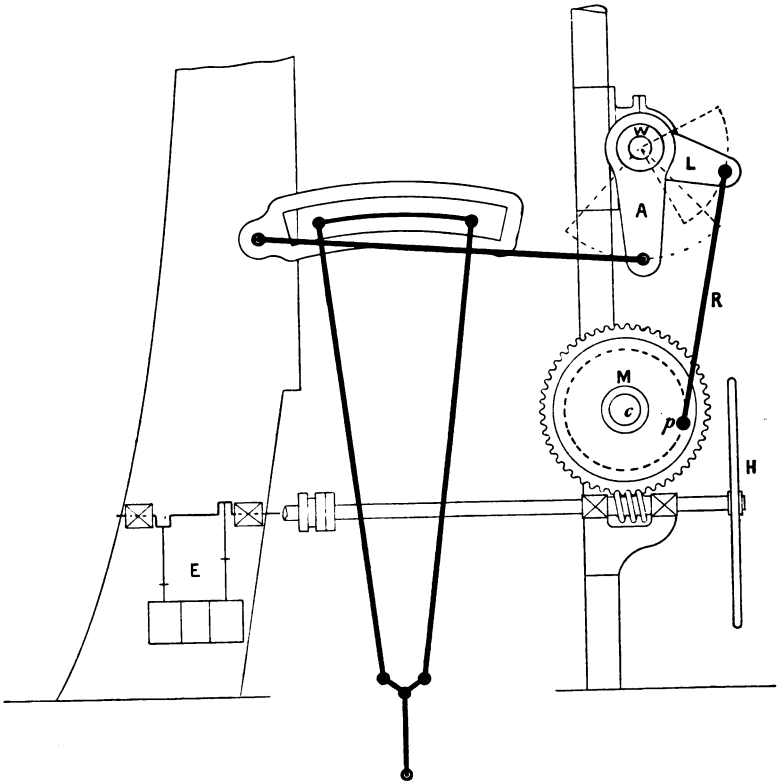


FIG. 77.

over; if the rotation of the wheel be continued the links will be brought back.

It will be seen that this "all-round" arrangement of the gear secures the links and fittings from injury from the cause referred to. Further, not only are automatic stopping arrangements unnecessary for the starting-engines, but these are no longer necessarily reversible, as was formerly the case; for, by rotation of the "all-round wheel" in one direction, the links may be carried into any desired position. There is, however, a distinct advantage in making the engines reversible even with this gear, and they are so fitted in recent engines for the service. In mercantile practice

the starting-engine, with the worm gearing by which it acts upon the weigh-shaft, is frequently replaced by a starting *cylinder*, the piston of which is connected directly with the reversing lever, so as to act upon it for turning the weigh-shaft by a simple push or pull. A hydraulic brake is fitted, as well as automatic stopping gear, to control the movement of the piston.

Summing up briefly the subject of link-motion and reversing gear: the link-motion affords a means of starting in either direction, and of stopping or reversing the engines, and also of working more expansively at reduced powers. These changes are produced by the simple movement of the links into different positions; and this is accomplished by means of the reversing gear, actuated either by steam or by hand.

Other Reversing Arrangements.—Special forms of valve gear are occasionally used instead of the link-motion, some using only one eccentric, others dispensing with eccentrics altogether; while for small auxiliary engines requiring reversal a simple *reversing valve*, by which the steam and exhaust communications may be interchanged by hand, is frequently used. These arrangements will not be further described here.

Special Expansive Gear.—In the older engines, in order to give the expansion required at reduced powers, earlier points of cut-off than are now used were necessary in the cylinders. These were produced by separate valves working independently of the slide-valves, and capable of being set to cut off the steam-supply to the slide-valves earlier or later in the stroke, within certain limits.

These *expansion valves* are to be found in the service in simple engines, and in the earlier double-expansion engines, but not in the later double-expansion nor in triple-expansion engines.

In compound and triple-expansion engines the expansion depends largely on the proportion of the cylinders; and a comparatively small variation in the point of cut-off is all that is necessary, such as may be approximated to by linking-up. An earlier cut-off than can well be produced by this means would be an undoubted advantage in these engines (with the proportions employed in the service) when working at reduced powers; but expansion valves are objectionable in several ways, and are very liable to give trouble.

In the later engines, therefore, linking-up is the only means available for increasing the expansion. The linking-up required, however, is not usually the same in the three cylinders; for

this reason means are provided in most engines by which each link may be independently adjusted, as follows:—

The suspension-rod end is attached to a block, *l*, Fig. 78, which traverses a slot, *s*, formed in the extremity of the reversing arm, A; and screw-gear is fitted for moving the block along the slot. The direction of the slot is such that, with the reversing arms set for ahead running, this movement will adjust the particular link; while, with the arms in the astern position the slots are perpendicular to the mean direction of the suspension-rods,

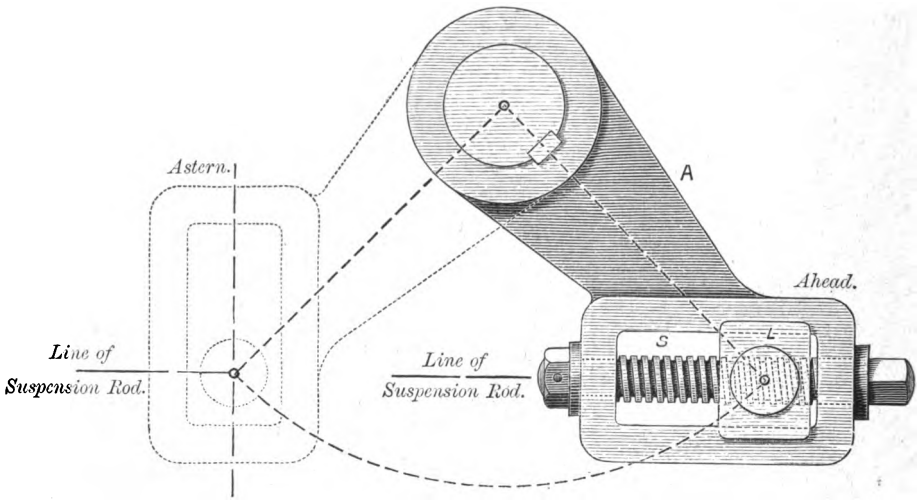


FIG. 78.

so that, whatever may be the positions of the blocks in the slots, the links are in full, or nearly full, astern gear. This ensures that the use of this *link-adjusting gear* for linking-up will not practically interfere with the ordinary action of the reversing gear in throwing all the links into astern gear at any moment.

Regulating-Valve (Fig. 79A).—This is the modern name of the throttle-valve, the object of which is to give immediate hand-control of the steam-supply to the engines. The valve is used in starting and in stopping the engines, and in regulating their speed. Its situation may be seen in Fig. 91, where the valve-box, R, is shown bolted directly to the slide-chest of the high-pressure cylinder.

As it has to be moved by hand under great pressure, a valve of the equilibrium type is used, the principle of which is shown

in the sketch Fig. 79, and the detail in Fig. 79A, the valve consisting of two disc valves, A and B, secured to the same valve-spindle, and opening or closing simultaneously two seatings by which steam admitted at C may pass to D. The steam has free access to the space, SS, between the outer and inner casings, and so to the outer surfaces of both A and B. The pressure is therefore practically balanced. By a hand-wheel and suitable gearing this double valve is worked from the starting-platform.

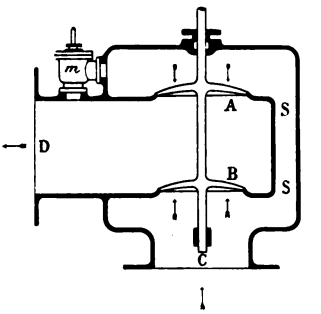


FIG. 79.

Manœuvring-Valve.—For making small alterations in the steam-supply, such as are required in manœuvring with a fleet, a small pass-valve, *m*, Figs. 79 and 79A, is fitted, by which a small amount of steam may be given independently of the regulating-valve.

Valves between Boilers and Cylinders.—All the valves on the main route from boilers to cylinders—viz. the stop-valves on boilers and bulkhead, the screw-down stop-valve, the regulating-valve, and the slide-valve—have now been considered. One kind of starting-valve has also been noticed in connection with the cylinder fittings. Starting-valves are more frequently simple pass-valves, used for giving steam by hand direct to the intermediate or the low-pressure slide-chests, as may be desirable in starting. Starting-valves are especially necessary in engines of modern type, in which only one of the cylinders receives the steam direct from the main steam-pipe, and consequently, in certain positions, prompt starting would be impracticable if steam could not be given independently to the other cylinders. The valves take steam from the boiler side of the regulating-valve box. They are worked by hand from the starting-platform.

Raising Steam, Starting, Under Weigh

Raising Steam.—It is highly desirable that steam should be raised very slowly in modern boilers, in order to avoid excessive strains from rapid and unequal expansion in the parts. With large boilers about eight hours are generally considered necessary for raising steam, and a longer time is often preferable. A good practice is to light the central furnaces, or one of the wing

furnaces, two or three hours earlier than the others, to set up a circulation and warm the water throughout. The difficulty is to displace the cold water from the bottom. A pump or other appliance is frequently used outside the service to withdraw this water and deliver it again to the boiler near the surface while raising steam. This is especially necessary when steam is to be raised quickly.

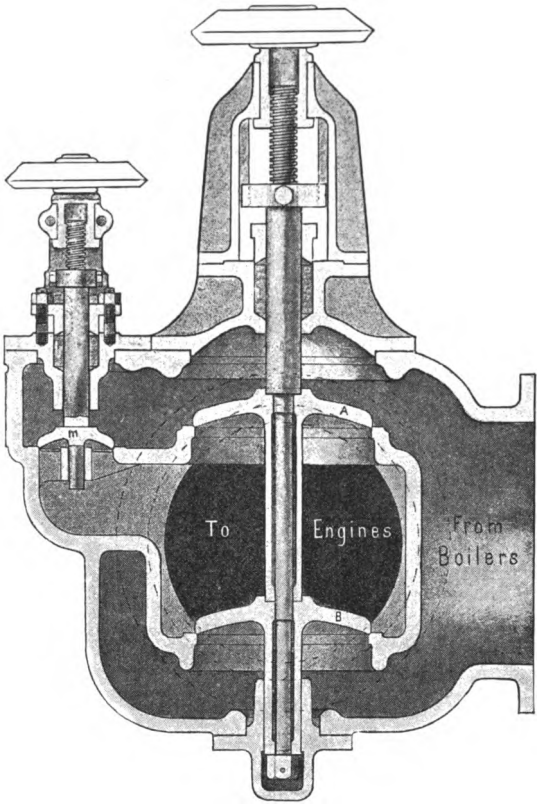


FIG. 79A.

Warming Engines.—It is important also that the cylinders, valve-chests, and all parts to which the steam will have access in working, should have been very gradually warmed up before starting, in order to avoid strains from sudden change of temperature, and also the condensation of steam on starting. The practice is to open all valves leading to the cylinders and cylinder-jackets as soon as the fires are lighted. The links are moved by hand

occasionally to allow vapour to pass to both ends of the several cylinders; and the cylinders, pistons, valve-chests, pipes, etc., are thus gradually warmed, the drains from these parts being freely used to get rid of the water formed by condensation.

As the pressure rises the regulating-valve is closed; but steam still has access to the jackets and to the starting-valves, and by means of these the warming of the cylinders is continued.

Starting.—When the engines are standing ready for starting, steam fills the main steam-pipe up to the regulating-valve, the links are in mid-position, the cylinders are warm and free from water, and steam is on to the jackets, to the starting-engine, and to the starting-valves. The centrifugal pump is at work circulating water through the condenser.

In order to start, it is necessary to run the links into gear for ahead or astern, and to open the regulating-valve. Should there be any difficulty, the starting-valves must be used as necessary. The engines should be started slowly, if possible, and the cylinder drains attended to.

Stopping and Reversing.—To stop the engines, steam is shut off, and the links are centered. In reversing, say from ahead to astern, steam is shut off, the links are then run over, and the regulating-valve reopened, the starting-valves being used if necessary.

Under Weigh at Reduced Speeds.—When under weigh, running at a given reduced power for a prolonged period, both boilers and engines are adjusted as far as possible for economical working.

The boilers generate steam most economically when working with a certain rate of combustion, usually about 15 lbs. of coal per square foot of grate per hour, or say 3 cwt. per hour per furnace, more or less, varying according to size of grate. This gives the number of fires, and therefore the number of boilers necessary. In the engines the expansion given to the steam should, as a rule, be as much as is practicable at the particular power. With the expansive gear so set, the pressure is kept at the point necessary for producing the power required, subject to the consideration that it must be sufficiently high for producing prompt reversal should occasion require.

Other prominent conditions for economical working are: careful management of the fires, proper use of the steam-jackets, correction of any sources of leakage, and maintenance of a good vacuum in the condensers.

CHAPTER VII

INDICATOR DIAGRAMS

AN Indicator Diagram from a steam-engine cylinder is a figure which represents, on a known scale, *the pressure acting on one face of the piston at each point of its path during a complete double stroke.* Such a figure is obtained by means of the *indicator*, Fig. 80, an instrument consisting essentially of a small cylinder *c*, fitted with an almost frictionless piston, and capable of being placed in communication at the lower end with the engine cylinder; while above the indicator piston, and connecting it with the fixed cover of the cylinder, is a spiral spring of known elasticity, by the resistance of which its movements under pressure are regulated. The space above the piston is in constant communication with the atmosphere.

The indicator may be fixed in a socket on the three-way cock which is fitted to the indicator pipe of each cylinder. By means of the cock the instrument may be put in communication with either end of the engine cylinder at will, or may be shut off from both ends. In addition, the instrument has its own stop-cock, *s*, at the lower part, by closing which the under side of the piston is shut off from the pipe and is put in communication with the atmosphere; the spring then assumes its normal length.

When the indicator is placed in communication with one end of the engine cylinder, with the engine at work, the varying pressure on the under side of the indicator piston causes it to rise and fall to an extent corresponding to the rise and fall of pressure in the engine cylinder, the compression or extension of the spring being proportional to the rise of pressure above, or to the fall of pressure below, the atmospheric pressure.

A pencil, *p*, moving with the indicator piston, is brought into

contact with a card, to which is given a transverse reciprocating motion corresponding to the motion of the engine piston on a small scale. The paper is fixed on a drum, *d*, within which is a coiled spring; and the motion is given by turning the drum through part of a revolution, during a stroke of the piston, by

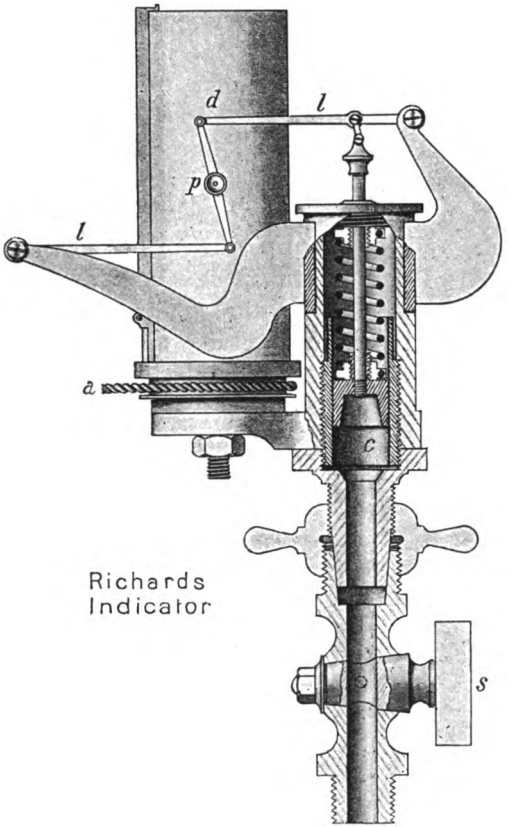


FIG. 80.

means of the cord and pulley, *a*, and allowing it to be returned by the spring during the return stroke. The motion is given from the piston-rod head so as to correspond to the motion of the piston.

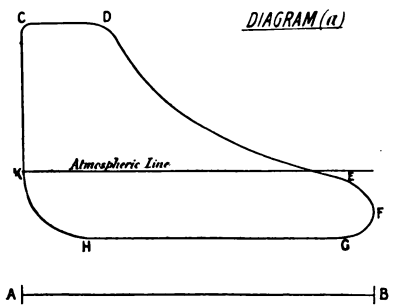
The pencil is kept on the paper during a revolution of the engine; that is, during a reciprocation of the card; and a diagram is thus drawn which registers the pressure at each point of the double stroke. A similar diagram being taken on the same paper

from the other end of the cylinder by turning the three-way cock and repeating the operation, a complete representation is obtained of the variation of pressure on both sides of the piston during a revolution.

The spring used above the indicator piston must be stiff enough to prevent undue vibration of the piston at high speeds. The movement of the piston is consequently very small, and the light levers, *l*, carrying the pencil at *p*, are for the purpose of magnifying this movement, in order to produce a diagram of reasonable height.

Diagram (*a*) shows the normal form of the diagrams obtained from a simple cylinder receiving steam for a portion of the stroke, and after expansion exhausting it to a condenser. The *atmospheric line* represents by its vertical position atmospheric pressure, and by its length the length of stroke; it is drawn with the stop-cock shut; when, by a small hole through the plug and shell, the under side of the piston communicates with the atmosphere.

The line AB below the diagram has been drawn merely for purposes of explanation; it represents the stroke of the engine piston. Suppose the indicator to be in connection with the end A of the cylinder, and the piston to arrive at A, and to be about to start again towards B when the pencil is brought into contact with the paper; at this moment the pencil rises to C under the action of the initial steam-pressure. As the piston moves forward the pencil first traces the line CD. At some point D the steam is cut off; after which, as the piston advances under the action of the expanding steam, the pencil falls, drawing the curved line DE. At point E release of the steam commences; and during the completion of the stroke the pencil, falling more rapidly, draws the line EF. The piston then commences the return stroke, and the pencil, continuing at first to fall, traces the line FGH. At H the exhaust is cut off for producing compression; and during compression the pencil rises, drawing the curve HK. At K the steam-port commences to open for admission of

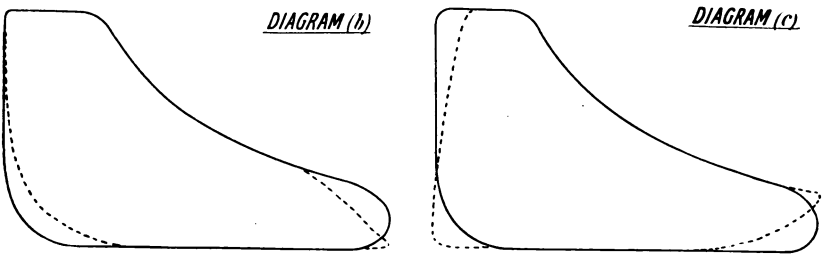


steam for the next stroke; and the pencil suddenly rises, drawing the line KC as the piston comes again to A. The figure is now complete, and the pencil may be removed from the paper. The names and some particulars of the several lines follow.

The *steam-line,* CD, indicates the pressure up to the point of cut-off; it usually falls more or less towards D.

The *expansion curve,* DE, shows the rate at which the pressure falls during expansion; this depends to some extent on the circumstances of the case, but, on the whole, the curve corresponds approximately with the law, *pressure varies inversely as the volume of the expanding steam.*

The *exhaust line,* EFG, shows the fall of pressure due to release; if this take place early the diagram may be modified as shown dotted in Diagram (*b*); if very late, as in Diagram (*c*).



The part GH of the *back-pressure line* indicates the back-pressure acting after the release period, while exhaust still continues. The back-pressure depends on the pressure in the condenser and on the freedom of exhaust.

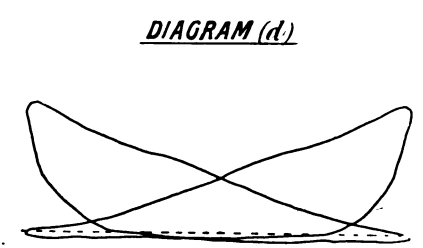
The *compression curve,* HK, indicates where compression commences, and the rise of pressure due to it; early and late compression are respectively shown dotted in Diagrams (*b*) and (*c*).

The *admission line,* KC, shows the rise of pressure due to admission; early and late admission are also shown in the above-named diagrams.

The conditions represented in Diagram (*c*) are bad; those in Diagram (*b*) are consistent with good working, though the valve operations take place earlier than usual. This result would be produced by large angular advance. For instance, diagrams taken when working linked-up exhibit these peculiarities, together with a falling steam-line and rising back-pressure line, due to the

contraction of the port openings to steam and to exhaust; see Diagram (*d*).

Compound Diagrams.—Fig. 81 illustrates the diagrams obtained from the three cylinders of triple-expansion engines. Such diagrams exhibit features which correspond with the special method of using the steam. A characteristic feature is the posi-



tion of the diagrams with reference to the atmospheric line. The high-pressure diagrams stand well above the line, the driving-pressure and the back-pressure being both high; the intermediate diagrams are lower, representing lower forward and back pressures; and the low-pressure diagrams are almost, or entirely, below the atmospheric line.

In these engines, as will be seen later on, the high-pressure cylinder receives steam from the main steam-pipe and exhausts

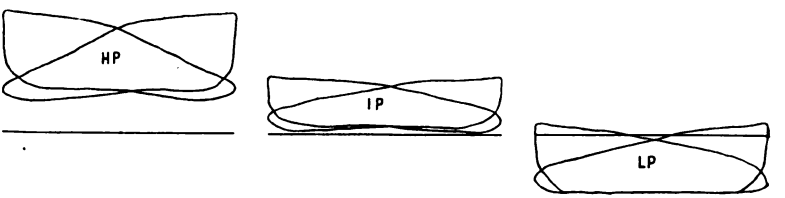


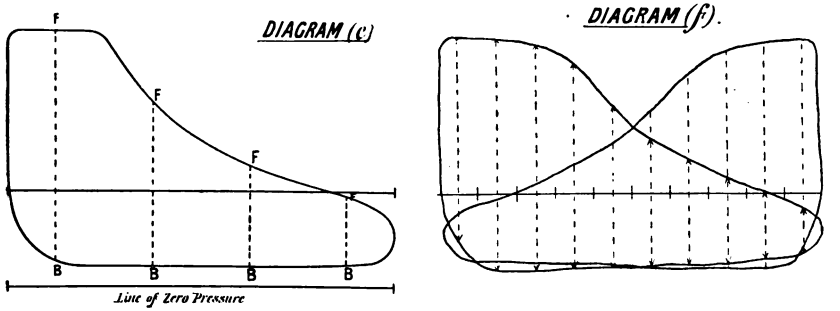
FIG. 81.

it to a receiver; the intermediate cylinder receives steam from this receiver and exhausts it to a second receiver; the low-pressure cylinder receives steam from the second receiver and exhausts it to the condenser. With the exception, therefore, of the steam-supply to the high-pressure cylinder, and the exhaust from the low-pressure cylinder, which take place as in a simple engine, the cylinders take steam from and exhaust to *receivers*, in which is a fluctuating pressure of steam; and this causes both the steam and the back-pressure lines, with the above exceptions, to be more or

less modified in form. Apart from this, however, the description of the simple diagrams previously given, as well as the whole of what follows with respect to measurement, will apply to these diagrams also.

Measurement for Mean Effective Pressure.—An important use of Indicator Diagrams is to enable the *mean effective pressure* exerted on the piston to be found, that is to say, the excess of the average driving-pressure over the average back-pressure. This quantity is an essential element in the calculation of the power.

In Diagram (*e*), vertical distances such as FB represent the excess of the steam-pressures during forward stroke over the back-pressures during return stroke, at the particular points of the stroke chosen. The actual pressures, whether forward or back, would be



given by the height of the points F or B above a line of zero pressure, placed at 15 lbs. below the atmospheric line; but it is unnecessary to find these pressures, the *difference* between forward and back pressure being alone required. By taking a sufficient number of such measurements as FB at equidistant points in the stroke, and finding the mean, we obtain the mean effective pressure for the diagram.

The usual method of finding this quantity is as follows:—The atmospheric line is divided into ten equal parts, and lines are drawn across the diagram perpendicular to the atmospheric line, and passing through points *midway between the points of division*, as in Diagram (*f*),—the principle being to measure, at its centre, the height of each of the ten slices into which the diagram would be divided if lines were drawn through the points of division.

Taking one diagram at a time, the lengths of the ordinates between the upper and lower lines of the diagram are measured,

either by an ordinary scale of inches or by a scale of lbs. pressure corresponding to the spring used in the indicator. The mean of the ten measurements so obtained is taken, and, if in inches, is converted into the corresponding number of lbs. pressure. This number is the mean effective pressure for the diagram measured. A similar process gives the number for the other diagram; and the mean of the two results is the mean effective pressure for the cylinder in lbs. per square inch.

The scale of pressures corresponding to the strength of the spring used is indicated by a fraction stamped upon it, which represents the fraction of an inch of compression or extension produced by each lb. of steam-pressure per square inch.

This scale is noted on the card, as well as the revolutions of the engines at the time the diagrams were taken. The steam and vacuum gauge indications are also usually noted.

CHAPTER VIII

PRINCIPLES—PROPERTIES OF STEAM

Work, Energy, Power

Work consists in overcoming resistance through space. This effect is the result of the exertion of force through space; in other words, of the exertion or expenditure of *energy*. The term energy is used to signify a *capacity for doing work*, from whatever cause the working capacity may arise.

In the steam-engine the force exerted is represented by the pressure of the steam upon the piston. Exertion of pressure is an inherent property of steam and other gaseous bodies, arising from their continual endeavour to expand under the influence of the heat-energy they possess.

Moving bodies obviously possess energy in virtue of their motion, since they are able to overcome more or less resistance before being brought to rest. This form of energy is known as *mechanical energy*. In the steam-engine, mechanical energy is developed in the moving parts of the machine by the action of the steam, and is applied by the machine to the performance of work.

Measurement of Work and of Energy.—Work is measured by the product of the *resistance* and the *distance through which the resistance is overcome*. Similarly, the energy exerted in doing work is measured by the product of the *force exerted* and the *distance through which it is exerted*. In either case the resistance or force is usually expressed, if in British Units, in pounds, and the distance in feet—the product then being in *foot-lbs.* The *foot-lb.*, which is thus the unit of work or of energy, represents the overcoming of *one lb. resistance through one foot distance*, or

the exertion of *one lb. force through one foot distance*, as the case may be.

In the steam-engine it is more convenient to measure the force exerted upon the pistons than it is to measure, directly, the resistances overcome; and *whenever the engine is running at a steady speed of revolution, the energy exerted on the piston during any number of complete revolutions is equal to the total work done in overcoming resistance in the same time.* In practice, therefore, it is the energy exerted or expended by the steam which is measured in order to find the work done.

In making this estimate, however, only the *effective* driving force must be taken into account, *i.e.* the force available for overcoming resistances after the resisting or *back* pressure of the exhaust steam has been allowed for, since the back-pressure must be balanced before any other result can be produced. The energy measured is therefore the *effective* energy, and is equal to the work done in overcoming all resistances except back-pressure.

Power.—In the case of a steam-engine, we require to know the *rate* of doing work; to express this the term *power* is used. If we say that an engine is working at the rate of so many foot-lbs. per minute, we are expressing the power of the engine's performance, but generally in very high numbers. For convenience, therefore, a large unit is commonly used in terms of which the power may be expressed; this is the *Horse Power*, which represents a performance of 33,000 *foot-lbs. per minute.* Thus if a resistance of 10,000 lbs. be overcome at the rate of 99 feet per minute, work is done at the rate of 990,000 *foot-lbs. per minute;* and this performance is represented by $\frac{990{,}000}{33{,}000}$, or 30 H.P.

Expression for I.H.P. developed in Cylinder of Steam-Engine.

If p = mean effective pressure on piston in lbs. per sq. in.,
= mean forward-pressure – mean back-pressure, found as already described,
and A = area of piston in square inches,
then $p \times A$ = *mean effective driving force, in lbs.*
If, again, L = length of stroke, in feet,
and N = number of strokes per minute (equal to 2 × revolutions),
then $L \times N$ = *distance through which driving force acts per minute, in feet.*
$\therefore$ $p \times A \times L \times N$ = *energy effectively exerted per minute, in foot-lbs.*

And this is the same as the effective work done per minute, when the engine is running steadily, as before explained.

$$\therefore \text{ H.P. developed in cylinder} = \frac{p \times A \times L \times N}{33{,}000}.$$

This is the actual, or Indicated Horse Power, so-called because the mean effective pressure is obtained from the indicator diagrams.

Example.—Find the I.H.P. developed in a cylinder, given mean effective pressure 20 lbs. per square inch, diameter cylinder 60″, stroke 4 feet, revolutions 85.

$$\text{I.H.P.} = \frac{p \times A \times L \times N}{33{,}000}$$

p = 20 lbs. per square inch,

A = area of piston, *in square inches* = $\pi \times 30^2$,

L = stroke, *in feet* = 4 feet,

N = strokes per minute = $2 \times 85 = 170$,

$$\therefore \text{ I.H.P.} = \frac{20 \times \frac{22}{7} \times 30 \times 30 \times 4 \times 170}{33{,}000} = 1165{\cdot}7$$

It is convenient to use $\frac{22}{7}$ for π in this calculation, since numerator and denominator may be at once divided by 11.

HEAT

The principal effects produced in substances by the application of heat are: *change of temperature, change of volume,* and *change of state*—as from the solid to the liquid state, or from the liquid to the gaseous.

Of these changes, change of temperature and change of state do not occur together. Thus, while a change of state is being produced by heat, the volume is generally undergoing change also, but the temperature remains unaltered. Heat which is thus employed in changing the condition of a substance without altering its temperature is called *latent heat,* while heat which produces, as one of its effects, a change of temperature is called *sensible heat.*

The question, which of the above changes will be produced in any given instance depends entirely on the conditions of the case; and the terms *sensible* and *latent* have no reference to any quality in the heat, but only to the effects produced by it.

Measurement of Heat.—Heat is usually measured as a

quantity by the change of temperature it is capable of producing in a given weight of water. The unit used in engineering in this country, known as the *British Thermal Unit*, is the quantity of heat necessary for raising the temperature of one lb. of pure water one degree Fahrenheit. This quantity is practically, though not strictly, the same at all temperatures. Thus, practically speaking, 100 B.T.U. will raise 100 lbs. of water through 1° Fahr. at any temperature; or, 10 lbs. through 10°, 1 lb. through 100°, and so on.

Mechanical Theory of Heat.—This is the accepted theory as to the nature of heat; it may be stated as follows:—

Heat is a form of energy, and is capable of transformation into other forms of energy; heat and mechanical energy are mutually convertible in the proportion of 1 B.T.U. to 772 foot-lbs. The theory is founded on the facts that heat and mechanical energy may each be produced by the expenditure of the other, and that (as was proved experimentally by Dr. Joule) in doing work against friction the quantity of heat produced is strictly in proportion to the mechanical energy expended.

The proportion is that given above; and the quantity 772 foot-lbs. of mechanical energy is therefore regarded as the equivalent of the unit of heat, and is called the *mechanical equivalent of heat.*

In accordance with this view, the work done in the steam-engine must be attributed to the conversion of heat into an equivalent quantity of mechanical energy.

Example.—An engine is developing 1000 I.H.P., what is the quantity of heat converted into mechanical energy per minute?

$$\text{Work done per minute} = 1000 \times 33{,}000 \text{ foot-lbs.},$$

$$\therefore \text{ Heat converted } \text{,,} = \frac{1000 \times 33{,}000}{772} = 42{,}746 \text{ B.T.U.}$$

STEAM

The ordinary operation of the steam-engine involves the following processes:—

Water is evaporated in the boiler under a certain pressure, which, when the engine is running steadily, is kept as constant as possible by management of the fires, the admission of steam to the

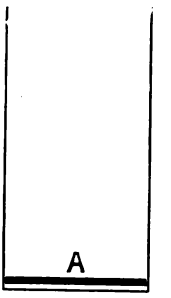
engines and the production of steam in the boilers proceeding at practically the same rate.

During more or less of the piston's stroke in either direction steam is freely admitted to the cylinder from the main steam-pipe; and work is done by the displacement of the piston, under the action of the steam-pressure, up to this point of the stroke. The steam is then cut off from communication with the boiler, and, thereafter expanding, drives the piston through the remainder of the stroke. The steam is then exhausted from the cylinder, while the piston is driven back by the action of fresh steam, admitted and allowed to expand in a similar manner on the other side of the piston. The steam exhausted passes to the condenser, either immediately as in simple engines, or after the expansive working has been continued in other cylinders, as in compound engines. In the condenser the steam is reconverted to water.

The principles involved in these processes will be briefly stated in a series of short notes.

Generation of Steam. NOTES 1-4

If water be placed at the bottom of an upright cylinder, beneath a loaded piston, A, the cylinder being open at the top, and heat be applied to the water until it is evaporated, the case will correspond to the practical one of evaporation in a steam-boiler under a constant pressure of steam in the steam-chest.



The effect of the continuous application of heat will be, first, to raise the temperature of the water to *some point at which boiling will commence,* which point will depend upon the pressure on the water; and, secondly, *at this temperature,* to convert the water gradually into steam having the same temperature but a greatly increased volume.

NOTE 1. *Pressure, Temperature, and Volume per Pound.*— In the process of generating steam, the pressure of the steam formed is equal to the pressure acting upon the water at the time of its formation; which pressure determines also both the temperature and the volume per lb. of the steam. Any given pressure carries with it during evaporation a certain *corresponding temperature, which is higher as the pressure is greater;* and also

a certain *volume per lb., which is smaller as the pressure is greater,* being, roughly, inversely proportional to the pressure.

Certain selected pressures, with the corresponding temperatures and volumes per lb., will be seen in columns 1, 2, 3 of the table following this note, which is given merely to illustrate the facts. The bracketed numbers in column 3 represent the proportion which the volume of the steam bears to the volume of the water from which it was formed.

In illustration of the foregoing remarks, supposing the water in the cylinder A to be 1″ deep, and the constant pressure upon it during evaporation to be, in three separate cases, 15 lbs., 75 lbs., and 170 lbs. per sq. in. respectively, the steam formed would fill the cylinder to the height of about 135 ft. at 15 lbs. pressure in the first case, 30 ft. at 75 lbs. pressure in the second, or 14 ft. at 170 lbs. pressure in the third; and the temperatures of the steam in the three cases would be 212°, 307°, and 368° Fahr. respectively.

EVAPORATION UNDER CONSTANT PRESSURE.

Gauge Press.	(1) Pressure (absolute).	(2) Temp.	(3) Volume per lb.	(4) (5) (6) Heat expended per lb. Steam (water originally at 100° F.)		
				Sensible.	Latent.	Total.
	Lbs. per sq. in.	Degrees Fahr.	Cubic Feet.	B.T.U.	B.T.U.	B.T.U.
0	14·7	212	26·36 (1643)	112	966	1078
15	30	250	13·5 (818)	150	939	1089
30	45	274	9·2 (577)	174	922	1096
60	75	307	5·7 (356)	207	898	1105
90	105	331	4·15 (259)	231	881	1112
120	135	350	3·3 (206)	250	867	1117
155	170	368	2·6 (165)	268	854	1122
185	200	381	2·25 (132)	281	844	1125

NOTE 2. *Condition of Steam.*—In its ordinary condition—that is, the condition in which it is produced and commonly used—steam is known as *saturated steam,* the name implying that it is a saturated vapour, or vapour having the maximum density consistent with its pressure or with its temperature. In another condition it is known as *superheated* steam. The ordinary or saturated condition is marked by the following characteristics:—

1. The steam possesses only as much heat as is absolutely necessary for its maintenance as steam at the particular pressure.

2. The temperature and the volume per lb. correspond with the pressure, as in the process of evaporation, being respectively

the *lowest temperature* and *the smallest volume per lb.* at which steam of the particular pressure can exist; in other words, saturated steam has the lowest temperature and the greatest density consistent with its pressure.

During formation, and so long as water is present with it, steam must be in the saturated condition just described; but when freed from the presence of water this is no longer a necessity.

Steam free from water, if supplied with heat, becomes *superheated;* in this condition the temperature and density of the steam no longer correspond with the pressure as during evaporation, the temperature being higher, and the volume per lb. greater, than the normal temperature and volume for the pressure.

For example, in the evaporating cylinder before referred to, heat supplied *after the evaporation is complete* would raise the temperature and increase the volume, while, from the circumstances of the case, the pressure would remain unaltered. The steam would become superheated, that is, it would acquire an amount of heat in excess of that possessed by ordinary steam of the same pressure; and it could then part with heat, until reduced to the ordinary or saturated condition, without any liquefaction occurring.

Superheated steam was used in low-pressure engines for increasing the efficiency of the expansion; but its use had to be abandoned with higher pressures, its high temperature and its dryness together causing great friction and wear and tear in the cylinders by rendering efficient lubrication impossible.

If ordinary steam carry moisture with it in a state of mechanical suspension it is called *wet* or *moist* steam; if free from moisture, *dry saturated* steam, or simply dry steam. Boilers rarely give steam in a perfectly dry condition. The application of heat to moist steam would first dry it by evaporating the moisture, and then, if continued, would superheat it.

Wet steam will become drier, or dry steam superheated, if it experience a fall of pressure without losing heat; for instance, if it suddenly expand to fill a larger space without doing work; the reason being that, at the lower pressure, less heat is required to maintain the steam in any given condition than at the higher. These effects are produced, for example, on exhaust from the cylinders to the receivers of triple-expansion or other compound engines.

NOTE 3. *Heat expended in producing Steam.*—This quantity consists, in all cases, of two parts: the *sensible heat*, viz. that used in raising the temperature before evaporation; and the *latent heat*, viz. that used in producing the actual evaporation.

These quantities are shown in columns 4 and 5 of the table.

It will be seen from column 5 that the latent heat necessary per lb. of steam is 966 B.T.U. at atmospheric pressure, but is less at higher pressures—the quantity diminishing with increased pressure by about $\frac{7}{10}$ B.T.U. for each degree the boiling-point is raised.

On the other hand, assuming any given temperature of the water originally, the quantity of sensible heat supplied per lb. will obviously be greater at the higher pressures than at the lower, since with the higher pressures the water must be raised through a greater number of degrees before evaporation commences.

The heat necessary for converting water of temperature W° into steam of temperature T°, under a constant pressure, is thus, *for each lb. of water*, approximately as follows:—

1st, *Sensible Heat*, for raising temperature from W° to T° = T − W units.
2nd, *Latent Heat*, for producing evaporation at temperature T° } = 966 − ·7(T − 212) units.

The sum of these quantities is the *Total Heat expended* per lb. of steam produced.[1]

Example.—A steady pressure of 100 lbs. absolute (corresponding temperature 328° Fahr.) is maintained in the boilers; the temperature of the feed-water is 110° Fahr.: find the total heat expended per lb. of steam.

Sensible heat per lb. water	= 328 − 110	=	218 B.T.U.
Latent heat „ „	= 966 − ·7(328 − 212)	=	885 B.T.U.
Total heat „ „	= 218 + 885	=	1103 B.T.U.

The total heat, reckoned from any given temperature of feed-water, increases only very slowly with the pressure, as may be seen from column 6. The practical result of this is that very little more fuel is required to convert water into high-pressure steam than would be required to convert the same water into low-pressure steam.

[1] In the expression for latent heat of evaporation, ·71 to ·72 is more accurate than ·7 for the higher pressures of, say, 100 to 200 lbs. per sq. inch; the expression as given, however, is sufficiently accurate for ordinary purposes.

NOTE 4. *Latent Heat of Evaporation: Effect of.*—The latent heat of evaporation is employed in doing work which is necessary both in the interior of the water and exterior to it, since the enlargement of volume produced is resisted internally by the molecular cohesion of the water, and externally by pressure. The overcoming of the external resistance is called the *external work* of evaporation; and this is the useful effect directly obtained.

Assuming a steady pressure maintained in the boiler, the external work of evaporation is represented by the work done on the piston while the cylinder is in communication with the boiler. This work is the equivalent only of a very small part of the latent heat expended, say $\frac{1}{10}$ to $\frac{1}{12}$; the remainder is necessary for internal changes, and represents the energy *acquired by the water* during evaporation. The total energy acquired by the water is represented by the sensible heat, plus the latent heat not spent in external work.

Condensation of Steam

NOTE 5.—Steam becomes readily condensed to water by contact with cold substances, contracting with great suddenness to the small volume it occupies as water, and leaving the space it previously filled practically, though not absolutely, free from pressure. This property is utilised in the steam-engine principally for reducing the *back-pressure* of the exhaust steam upon the piston, which pressure would not be less than about 18 lbs. per square inch if the steam were exhausted to the atmosphere.

In marine engines the steam is exhausted to a condenser. This is a closed vessel, practically exhausted of air by an air-pump, and kept cold enough by means of cold water to instantly condense the steam which enters it. Under these circumstances there exists within the condenser only a very low pressure, due in part to uncondensed vapour, and in part to air, neither of which can be completely removed; and the pressure on the exhaust side of the piston approximates to this low condenser pressure. The actual pressure in good condensers is commonly about $1\frac{1}{2}$ lbs. per square inch, and the back-pressure in the cylinder about 3 lbs. The condenser, therefore, practically increases the effective driving pressure in the cylinder which exhausts to it by about 15 lbs. per square inch, and thus adds both to the power and economy of the engine.

Other advantages are that the exhaust is a silent one, and that warm fresh feed-water (assuming a surface-condenser) is available for the boilers.

The *vacuum-gauge*, constructed on the same principle as the pressure-gauge previously described, shows, in inches of mercury, the difference between the actual pressure within the condenser and the atmospheric pressure. A good vacuum would be indicated by about $26\frac{1}{2}''$ to $27\frac{1}{2}''$. The difference between the gauge reading and the barometer reading, divided by 2, will give the actual condenser pressure in lbs. per square inch.

Action of Steam. NOTES 6-11

NOTE 6. *Expression for Work done in driving Piston.*—The work done in driving a piston by fluid pressure, as in steam, air, or hydraulic engines, is *proportional to the product: average pressure* × *volume through which piston is driven.*

This is obvious, since—

Work done = average pressure × area piston × distance piston is moved through; and, area piston × distance moved through is volume moved through; proper units being taken in both cases.

The actual relation is—

Foot-lbs. Lbs. per square inch. Cubic feet.

$$\textit{Work done} = \textit{average pressure} \times \textit{volume moved through} \times 144.$$

Thus, if 20 cubic feet of water be admitted to a hydraulic press at a pressure of 500 lbs. per square inch, the work done by it is $500 \times 20 \times 144 = 1{,}440{,}000$ foot-lbs., assuming the whole of the water to be admitted to the space through which the piston or ram actually moves; in other words, assuming that no clearance spaces have to be filled.

NOTE 7. *Expansive Property of Steam.*—In a hydraulic or water-pressure engine, in which the working fluid is non-expansive, pressure will be exerted on the piston only so long as the communication remains open between the cylinder and the water-supply. Immediately the communication is closed the pressure will cease, and the piston will have been moved only through a volume equal to the volume of water admitted.

In the steam-engine the case is different, owing to the

property of expansibility which steam possesses in common with all vapours and gases, in virtue of which they tend to expand indefinitely.

The steam used works as follows:—

1. From the commencement of the stroke to the point of cut-off, the steam entering the cylinder displaces the piston through its own volume, with a certain pressure. The action so far corresponds with that of the water-pressure engine.

2. The steam in the cylinder being cut off from the source of supply, the expansive property is able to take effect. The steam will expand if its pressure be sufficient to overcome the resistance; and will continue to expand, if allowed, so long as this is the case. After cut-off, therefore, the piston is still driven, though with a gradually diminishing pressure (Note 9); and additional work is thus done without further expenditure of steam.

NOTE 8. *Admission Work per Lb. of Steam Practically Constant.—Expansive Work the Source of Economy.*—The work done may thus be divided into two parts, viz. *admission work* and *expansive work*. The admission work is the result of the increase of volume which occurs during evaporation, and it represents the external work of evaporation previously referred to (Note 4); the expansive work is the result of subsequent increase of volume, and is done at the expense of heat-energy possessed by the expanding steam.

The amount of admission work done per lb. of steam is practically the same at all pressures; being in proportion to the product of pressure and volume (Note 6), which is practically a constant quantity (Note 1). This fact shows that high-pressure steam would be no more economical than low-pressure steam if both were used without expansion, both the fuel burnt (Note 3) and the work done being practically the same in the two cases for a given weight of steam used.

The economy of high-pressure steam arises from its greater capacity for expansion; and the characteristic feature of modern steam-engines consists in the large extent to which expansive working is employed.

NOTE 9. *Approximate Law of Expansion.*—During the expansion of steam its pressure falls continuously, and in accordance, approximately, with the law: *pressure varies*

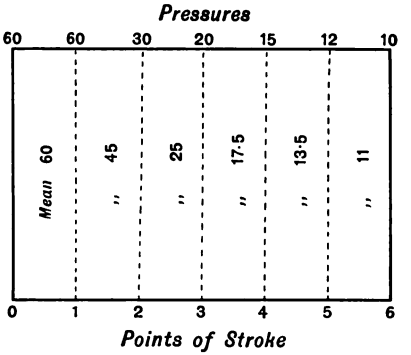
inversely as volume of expanding steam. For instance, if steam of volume A expand to volume B, then

$$\frac{\text{pressure at volume B}}{\text{pressure at volume A}} = \frac{\text{volume A}}{\text{volume B}}.$$

The following example will serve for illustration :—

Example.—Steam of 60 lbs. pressure (absolute) is cut off at $\frac{1}{6}$ stroke ; find, approximately, the average pressure during the stroke.

Divide the stroke into six equal parts, as in the figure.



The expansion commences at point 1, where volume is represented by 1, and pressure is 60 lbs.

Then, at point 2, where volume = 2, $\frac{\text{pressure}}{60} = \frac{1}{2}$, or pressure = $\frac{60}{2}$

„ „ „ 3 „ „ = 3, $\frac{\text{pressure}}{60} = \frac{1}{3}$, or pressure = $\frac{60}{3}$,

and so on.

These pressures are shown above the figure. The numbers shown in the six spaces are the average pressures for those spaces. Thus, between points 2 and 3 the average is $\frac{30 + 20}{2} = 25$.

Finally, taking the mean of these averages—

$$\text{Average pressure} = \frac{60 + 45 + 25 + 17{\cdot}5 + 12{\cdot}5 + 11}{6} = 28{\cdot}7 \text{ lbs. per square inch.}$$

The law would have been more correctly applied if the clearance spaces of the cylinder had been taken into account. At the moment of cut-off in the above example, steam fills not

only one-sixth of the cylinder, as here supposed, but also certain additional spaces, consisting of the clearance at the end of the stroke, and the steam-passage; consequently the steam actually expanding is more than that considered, and the actual pressures will be somewhat higher than those shown, especially towards the end of the expansion. The calculation, however, is not of much practical importance.

NOTE 10. *Theoretical Indicator Diagram.*—Fig. 82 shows the theoretical indicator diagram corresponding to the above example.

The base-line representing the stroke is divided as before. Ordinates are erected proportional in height to the pressures at the various points, and their upper extremities are joined by a line which forms a steam-line and an expansion-curve.

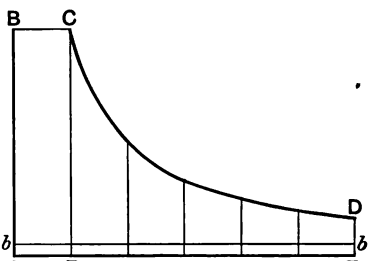


FIG. 82.

Such a figure is useful for representing the theoretical action of the steam. The *area of the diagram* represents on some scale the *work done*, since its average height represents average pressure and its length is proportional to volume. If a line, *bb*, be drawn at a height corresponding to the assumed back-pressure, say 3 lbs., the area above it represents the *effective work*.

NOTE 11. *Benefit of Expansion.*—Referring to the above diagram, the area ABCF represents the *admission work*; in the example it is proportional to the product 60×1. This is all that could be obtained without expansion from the quantity of steam used.

The area FCDE represents the expansive work, and ABCDE the total work. In the example the total work is represented on the same scale by the product $28{\cdot}7 \times 6 = 172$.

Thus the work theoretically due to a given quantity of steam is in this case nearly three times as much as when used without expansion; and the *effective* work is increased in nearly the same proportion.

The theoretical gain by expansion may be further illustrated thus:—

Let steam of say 10 lbs. pressure be admitted to a cylinder

for the entire stroke AB, Fig. 83; the total work is represented by the area of the rectangle A, 10, 10, B, using the numbers merely as reference letters. Next, let the steam be admitted at 20 lbs. pressure and cut off at $\frac{1}{2}$ stroke; the *quantity of steam used* is the same as before, because after expansion it will fill the same cylinder at the same pressure, viz. 10 lbs., but the work is now represented by the area A, 20, 20, 10, B. Going to 50 lbs. pressure, and cutting off at $\frac{1}{5}$ stroke, the work area becomes A, 50, 50, 10, B. With 100 lbs. pressure and cut-off at $\frac{1}{10}$ stroke, the work area will be A, 100, 100, 10, B. The quantity of steam used is the same in all these cases.

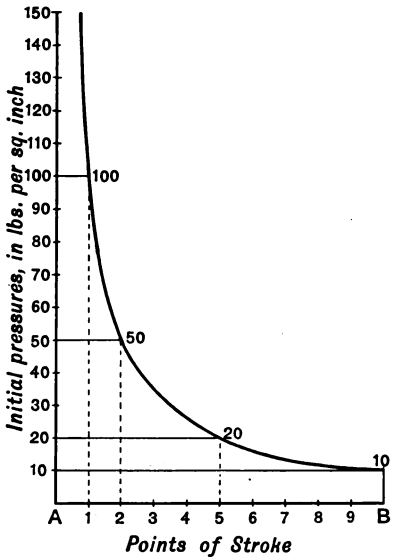


FIG. 83.

This process might be continued indefinitely; and it will be seen that as the expansion increases, so also does the work theoretically obtainable from a given quantity of steam (a given weight of water evaporated), though the further gain becomes less and less for a given increase in the expansion. It is clear, too, that great expansion involves a high initial pressure in order that the final pressure may not be reduced to too low a point.

Efficiency

NOTE 12.—The water used in the steam-engine is merely a convenient medium for the performance of work by the action of heat. Supplied to the boiler, say from the condenser, it receives heat, and undergoes great expansion both during evaporation and subsequently in the cylinders; exhausted to the condenser, it gives up heat and undergoes great contraction, returning to its original condition and temperature. The result of these changes of volume, and of the accompanying pressures, is that work is done on the pistons. This work is the equivalent of only a small part of the

heat spent upon the steam; the remainder being lost, chiefly by the unavoidable discharge of the exhaust steam. The fraction of the heat-expenditure which appears as effective work represents the *efficiency* of the steam (or of the engine as a means of converting heat into work); thus—

$$\text{Efficiency of steam} = \frac{\text{Effective Work (in heat units)}}{\text{Total heat expended on steam}}.$$

This fraction, in the best modern engines, is about $\frac{1}{6}$.

It must not be supposed, however, that if all imperfections in the engines could be removed, the whole heat spent might be converted into work. Investigation shows that the efficiency attainable would still be limited, and would depend upon the range between the highest and the lowest temperatures of the working fluid.

On this point a simple statement will suffice. If the temperature of the steam in the boiler, and of the condensed steam in the condenser, be $t_1°$ and $t_2°$ Fahr. respectively, the efficiency of the *theoretically perfect engine* would be given by the fraction—

$$\frac{t_1 - t_2}{t_1 + 461}.$$

These temperatures being, in modern engines, about 370° to 380°, and 100° to 110°, the limiting or maximum efficiency theoretically attainable is rather less than $\frac{1}{3}$, instead of the $\frac{1}{6}$ practically attained. The ideally perfect engine involves impracticable conditions of working; it is, however, instructive to have the means of comparing the actual efficiency attained with the theoretical limit of efficiency.

Example on Efficiency.—An engine uses 15 lbs. of steam per hour per I.H.P. developed. The temperature of feed-water from condenser is 106° Fahr.; the temperature of boiler steam 362° Fahr. Find the efficiency of the steam.

Heat spent, per lb. steam = $362 - 106 + 966 - \cdot7(362 - 212)$
$= 1117$ B.T.U.

$\therefore$ Heat spent per hour, per I.H.P. $= 15 \times 1117 = 16{,}755$ B.T.U.

Work done per hour, per I.H.P. $= 33{,}000 \times 60 = 1{,}980{,}000$ ft.-lbs.

$\therefore$ Efficiency of steam $= \dfrac{1{,}980{,}000}{772 \times 16{,}755} = \cdot153$, or $\dfrac{1}{6\cdot5}$ approximately.

CHAPTER IX

EXPANSIVE WORKING IN PRACTICE

THE great increase in the working pressure of steam since the early days of the marine engine has already been referred to. The principal object has been fuel economy, the importance of which has steadily increased with the increase of the propelling power provided.

The engines which have been used with the various pressures, classified according to their method of using the steam, are as follows :—

1. **Single-expansion** engines (otherwise simple engines), with boiler pressures up to 35 lbs.

2. **Double-expansion** engines, commonly known as compound engines (otherwise bi-compound), with pressures from 60 lbs. to about 100 lbs.

3. **Triple-expansion** engines (otherwise tri-compound), the present type of compound engine, with pressures ranging from 120 lbs. to 170 lbs. or more.

Single-expansion or Simple Engine (Fig. 84). — The two cylinders, A, A (assuming a pair of engines), are of the same size, and steam is supplied to each of them independently. One-half the steam is received by each cylinder, is cut off at some point of the stroke, and at the end of the stroke is exhausted directly to the condenser. The entire expansion is therefore given during a single stroke; and it is obvious that large expansion can only be produced by an early cut-off.

Double-expansion Engine (Fig. 84).—Here the cylinders A and B have the same stroke, but are of unequal diameters; the volume of the *low-pressure cylinder* B being three to four times the volume of the *high-pressure cylinder* A. Steam from the boiler

is directly supplied only to A, where it works during a stroke with a very moderate expansion. Exhausted from A, the steam passes into B during a stroke in that cylinder, driving the piston and expanding meanwhile in consequence of the larger volume into which it is passing. At the end of the stroke in B the steam is exhausted to the condenser. The expansive work begun in A is thus continued in B; and the total expansion would be found by multiplying the expansion given in A by the ratio of B to A, seeing that the steam cut off in A fills B just before final exhaust.

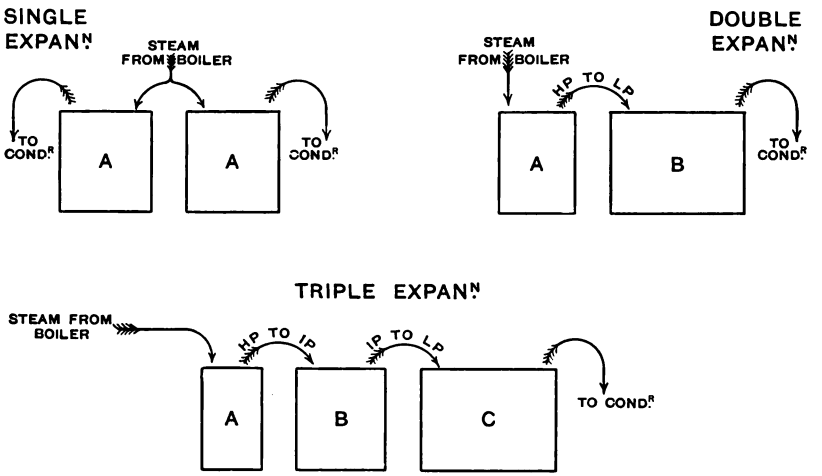


FIG. 84.

Arrangements of this type of engine are shown in Figs. 85, 86, and 87, and will be referred to later.

Triple-expansion Engine (Fig. 84). — This is merely an extension of the last type. In the simplest form there are three cylinders, driving three cranks. The cylinders have the same stroke, but are of unequal diameters: the smallest, A, being the high-pressure; the largest, C, the low-pressure; and B the intermediate cylinder. Steam is supplied direct to A, and, after a stroke in which it receives a very moderate expansion by "cut-off," it passes to B during a stroke in that cylinder, and then to C during a stroke in that cylinder, at the end of which it is exhausted to the condenser.

The expansive work begun in A is continued first in B and later in C; and the total expansion is now the expansion given in

A multiplied by the ratio of C to A, since the steam cut off in A fills C just before final exhaust. The volume of C is five to seven times the volume of A. This point will be referred to later.

Comparing the above types, it will be seen that in the simple engine the expansion is the result of cut-off alone; while in the compound types it is the result also of inequality in the sizes of the cylinders, and of these being used in succession. The essential distinction is that, whereas in the simple type the total expansion required must be given in each of the cylinders, in the compound types only a part of the expansion is so given.

Now experience has shown that the latter system is more economical than the former; and that, *while large expansion is necessary for securing economy, only a very limited amount of it ought to be given in any one cylinder.*

The principal reasons for this practical limitation are as follows. Steam cylinders are exposed on their internal surfaces to a varying temperature, ranging from that of the entering steam to that of exhaust. This would necessarily involve some waste of heat, the surfaces becoming more or less cooled during exhaust, and abstracting heat from the entering steam during admission by the ordinary processes of radiation and conduction; but the actual waste is liable to be very much greater than would be caused by this action on account of the presence of moisture.

Moisture is deposited on the internal surfaces during admission, by condensation of steam on contact with the previously cooled metal; and, when exhaust occurs, moisture lying on the metal readily evaporates under the reduced pressure, taking heat from the surfaces and leaving them greatly cooled. These surfaces are reheated during each admission period at the expense of the entering steam, only to be recooled, as described, during exhaust. A complete account of the action referred to would be somewhat complicated, for during the expansion of the steam both deposition and evaporation of moisture occur; but the condensation during admission and the evaporation of moisture during exhaust are the most important features. If a quantity of water be supposed to remain in the cylinder during the working of the engines it will act in the same way as the cylinder metal; and in general, probably, the waste is partly due to this cause.

The action of the cylinder metal, and, perhaps, of water, in thus receiving heat from the steam and employing it in useless evaporation, constitutes a serious waste. This waste becomes greater with increased expansion in the cylinder (the range of temperature and the amount of moisture formed being both greater with increased expansion); and it seriously detracts from, and may entirely annul, the advantage otherwise due to the expansion.

The waste just described is checked by the action of the steam-jacket, and is largely reduced by the use of the compound system of expansion.

Steam-Jacket.—The office of the steam-jacket is to supply heat from without to the inner surface of the cylinder metal, in order to check the deposition of moisture on the metal. More or less of the liquefaction which would have occurred within the cylinder, causing there a greatly increased cooling action during exhaust, is transferred to the jacket, where there is no such action, and where the presence of water does not lead to waste, but merely necessitates occasional draining. The action of steam-jackets, however, though certainly beneficial, is limited in extent; so that the expansion in the cylinder still requires to be moderate.

Compound Expansion. — The principle of the compound system of expansion may be thus briefly stated. The large expansion desired for the steam is obtained, not by cutting it off early in the stroke, but by using it successively in two, three, or more cylinders, in each of which *one stage* of the expansion is carried out, each stage representing only a moderate expansion. The effect is that the waste by cooling described above, which largely depends on the amount of expansion given in individual cylinders, is much reduced. In the double-expansion engine two stages are used; in the triple-expansion, three stages; in the quadruple-expansion engine, of which some examples are at work, four stages.

The compound system conduces not only to economy, but also to smoothness of working. Large expansion given during a stroke would produce increased irregularity of driving and increased stress on the machinery for a given power, the maximum pressure being greatly in excess of the mean. This excessive irregularity is avoided by the compound method.

Arrangements of Compound-expansion Engines

Double-expansion Engine.—The three arrangements of this engine in use in the Navy, viz. the *Woolf* or *Tandem* type, the *Two-cylinder side-by-side* type, and the *Three-cylinder side-by-side* type, are illustrated by simple sketches, Figs. 85, 86, and 87. To all these the description already given of the working of the double-expansion engine will apply. High-pressure cylinders are marked A, low-pressure cylinders B.

In the *Woolf* type (Fig. 85) each engine has two cylinders—one high-pressure, one low-pressure. These are placed in line one

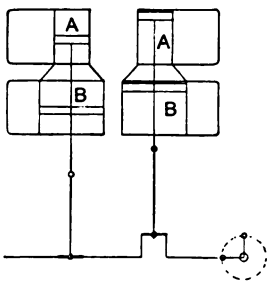


FIG. 85.

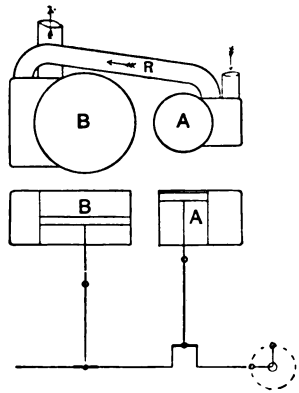


FIG. 86.

behind the other, and the two pistons work together, being on the same rod, which is prolonged sufficiently to carry both. The cranks of the two engines are placed at right angles as usual. Steam is supplied direct to the two high-pressure cylinders; thus half the supply passes through the cylinders of each engine, receiving expansion in two stages.

In the *Two-cylinder* type (Fig. 86) the cylinder of one engine is the H.P. cylinder; the cylinder of the other is the L.P. cylinder. The cranks of the two engines are at right angles as usual, for facility in starting and smoothness of turning; consequently the high and low pressure pistons do not make their strokes simultaneously, and the steam does not pass directly from H.P. to L.P. cylinder during a stroke of the engine, as in the last type, there being now a marked interval between exhaust on the high and

admission to the low. This fact, however, is not of any importance so far as the final effect is concerned.

The pipe R, shown in the upper view, Fig. 86, to which the H.P. exhaust takes place, and from which the L.P. cylinder takes its steam, is called the *receiver*; this term should be understood as including also the spaces which the pipe connects, viz. the exhaust passage at the one end and the slide casing at the other. The course of the steam is shown by the arrows.

In comparison with the tandem type, this type requires only half the number of cylinders; but they are necessarily of larger dimensions, since in each case the cylinder must be equal in capacity to the *two* similar cylinders of the other type. Thus the low-pressure cylinder becomes very large in powerful engines of this type. For this reason it is preferable, in the largest powers, to employ *two low-pressure* cylinders in connection with one high-pressure cylinder; this brings us to the *Three-cylinder* type, shown in Fig. 87. In this arrangement the three cylinders work three separate cranks, which are usually placed at 120° apart. The receiver R, shown in the upper view, connects the H.P. exhaust passage with the two L.P. slide-valve chests; so that the steam supplied to the high is divided on exhaust between the two low pressure cylinders as shown by the arrows (though not quite equally, since one takes steam before the other), and from them is exhausted to the condenser, having undergone expansion in *two stages* as before.

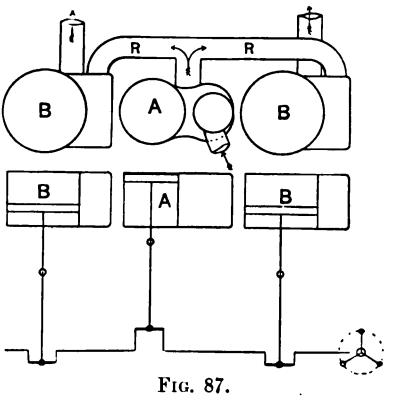


FIG. 87.

Comparing the three types of double-expansion engine, though there is little difference in point of economy between them, the two-cylinder type has been the favourite form and the most widely fitted for moderate powers, as being the simplest in construction and having the fewest parts. The three-cylinder type has been largely fitted for the higher powers, and is to be found principally in armoured vessels, from the date of the *Alexandra* down to and including the *Admiral* class of battle-ships; it has therefore usually been a vertical engine, while the two-

cylinder type has usually been horizontal. The three cranks, while involving an increased number of working parts, and greater length of engines fore-and-aft, have compensating advantages, one of which is the greater uniformity of turning. The tandem type has not been widely fitted in the service; it has the great disadvantage of requiring increased length in the direction of the stroke, thus increasing the difficulty already felt in regard to space in that direction.

Triple-expansion Engine.—The most important arrangement of this engine, and the only one used in the service, is that which has already been briefly described, in which three cylinders, A, B, C, Fig. 88, all of different diameters, and placed side-by-side, work three cranks; and the steam, supplied direct to the smallest, passes through them in succession to the largest, undergoing expansion in three successive stages.

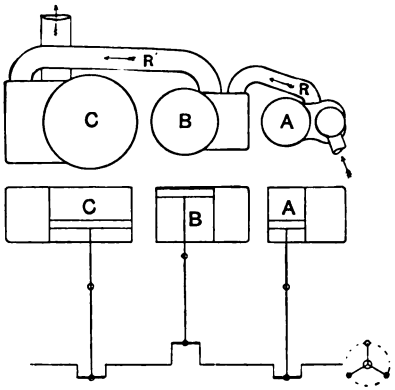


FIG. 88.

In the upper view, Fig. 88, are seen the pipes by which the steam passes, viz. the main steam-pipe leading to the high-pressure cylinder, the receivers between the cylinders, and the exhaust-pipe from the low-pressure cylinder. The *first receiver*, R, connects the exhaust orifice of the high-pressure with the slide-chest of the intermediate cylinder, for which cylinder it acts as the steam supply-pipe; the *second receiver*, R′, similarly connects the intermediate and the low-pressure cylinders. The course of the steam is shown by the arrows. The cranks are placed at 120° apart; the pistons therefore do not make their strokes simultaneously.

A longitudinal section of a set of triple-expansion engines is given at the commencement of the book.

Proportion of Cylinders.—The proportion of low-pressure to high-pressure cylinder capacity in triple-expansion engines outside the service ranges from about $\frac{6}{1}$ to $\frac{7\cdot5}{1}$ for working pressures of 140 to 175 lbs. With 155 lbs. working pressure it is about $6\frac{1}{2}$ or $6\frac{3}{4}$ to one. A cut-off at say $\frac{1}{2}$ stroke in the H.P. cylinder will thus produce a high degree of expansion.

In service engines a smaller ratio, viz. 5 to 1, is usual (the three cylinders being as 1 : $2\frac{1}{4}$: 5), and also a later cut-off at full power, viz. about $\frac{3}{4}$ stroke; ranging to about $\frac{1}{2}$ stroke or a little earlier at reduced powers; the expansion given is therefore smaller, especially at full power.

The following are the reasons for this difference:—(1st) In the Navy it is of very great importance to keep down the size and weight of the engines in proportion to the full power developed. Moderation in the expansion conduces to this end by reducing the sizes of the cylinders, with their pistons, slide-valves, and casings. (2nd) The engines of a war-ship should work as economically as possible through a great range of power. For instance, the power usually required is only about 10 to 15 per cent of the full power which they are designed to develop. No set of engines can be designed to work with real efficiency at such widely different powers. The large cylinder capacities suitable for efficient working at the highest power would be much too great for efficient working at the low powers generally required. The more moderate capacity of cylinder actually used is more favourable for efficiency at reduced powers, though still so large that, in order to reduce the power sufficiently, it becomes necessary to greatly reduce the initial pressure, although expanding as much as possible. The engines are therefore at once too small for the highest powers, and too large for the lowest, to obtain the highest degree of efficiency at either of those powers; and even the intermediate powers are interfered with by the necessity of arranging for the extreme conditions. The boilers also at full power are very much pressed. The most efficient powers, tested by the consumption of coal per I.H.P., are generally those between 40 and 60 per cent of full power.

In merchant and passenger vessels the power intended to be regularly used at sea is nearly the full power, and the engines and boilers are designed accordingly, *i.e.* the *ordinary power* is developed with full pressure and expansion, and with boiler and cylinder capacities suitable for economical working at that power. The difference between the two cases is obvious; and the facts stated account for a smaller degree of economy being attained in naval engines.

Receiver Pressure.—During the working of the engines the receivers contain a quantity of steam which is added to and

drawn upon periodically during the revolution, and which varies in amount according to the circumstances of working. On starting the engines, steam accumulates in the receiver until a certain average pressure is reached, after which *as much steam passes out of the receiver per stroke as is discharged into it.* The receiver pressure necessarily adjusts itself to the point of cut-off in the receiving cylinder so as to produce this result. If, for example, we cut off in the receiving cylinder earlier, thus providing less space in it for the reception of the steam, the pressure in the receiver will rise, by reason of some steam which it receives being left behind at each stroke, until the above equality is again established. Similarly, if we cut off later, the receiver pressure will fall correspondingly. Though the term *receiver pressure* is used, the actual pressure is obviously a fluctuating one, rising and falling during each revolution as the cylinders discharge steam to, or take steam from the receiver, and as the pistons move to diminish or increase the cylinder spaces in connection with the receiver.

Effect of Receiver Pressure, and of altering Points of Cut-off.—The receiver pressure obviously acts as a back-pressure in the exhausting cylinder. For instance, the pressure in the first receiver acts as the back-pressure in the H.P. cylinder, and reduces the effective work in that cylinder. This, however, does not mean loss of work finally, but simply that some work which would have been done in A is transferred to B. Similarly, the pressure of the second receiver acts to transfer work from B to C. The higher these receiver pressures are, the greater is the work transferred from the earlier to the later cylinders, and *vice versa;* and here we have a means of regulating the division of the total power between the three engines. If, for example, leaving other things the same, we cut off a little earlier in the intermediate cylinder, the effect is to raise the pressure in the first receiver; this increases the back-pressure on the H.P. piston and the forward-pressure on the intermediate piston, thus causing more work to be done in the intermediate, and less in the high-pressure cylinder, while the total work of the two engines remains practically the same if the alteration be a small one. An earlier cut-off in the L.P. cylinder would produce a similar change in the division of power between that engine and the intermediate.

The effect of altering the point of cut-off in the H.P. cylinder

is, with any given working pressure: (1) to alter the amount of steam passed through the engines per revolution, and, consequently, the mean pressures, the revolutions, and *the total power developed;* (2) to alter the expansion given in the H.P. cylinder, and consequently *the total expansion.* For instance, cutting off *earlier* in the H.P. cylinder gives reduced power with increased expansion.

Expansion not wholly effective—"Drop."—The expansion has been hitherto supposed to be a *continuous working expansion;* but in compound engines this is not altogether the case, more or less of the expansion taking place on exhaust to the receivers without useful effect, so that the continuity of the working expansion is interrupted by gaps occurring between the cylinders. There is, in fact, more or less fall, or "drop," of pressure between the cylinders from unbalanced expansion. This arises from the necessity for dividing the power fairly equally, as will be seen from what follows.

Calling the cylinders as before A, B, C, then, in order that the expansion should be continued unbroken, say from A to B, the volume up to cut-off in the cylinder B would require to be equal to the volume of A. For instance, if the volume of B be double that of A, then the cut-off in B must be at $\frac{1}{2}$ stroke; if three times, then at $\frac{1}{3}$ stroke; and similarly as between B and C.

Thus, with the actual proportions of cylinders used, the cut-off in the I.P. and L.P. cylinders would require to be before $\frac{1}{2}$ stroke; and the high receiver pressures so produced would transfer too much of the work to the later cylinders, especially at high powers.

The diagram A, Fig. 89, illustrates this point. The volumes of the cylinders are $1 : 2\frac{1}{4} : 5$, as shown on the base line: the cut-off in the intermediate gives a volume = 1 for receiving the steam, and in the low a volume = $2\frac{1}{4}$; the expansion is therefore continuous, but the areas of the diagrams show that too little of the work is done in the H.P. cylinder and too much in the L.P. To reduce this inequality it is necessary to cut off at later points than those above indicated, so as to lower the receiver pressures, and thus allow the H.P. cylinder to do a larger share of the work and the L.P. cylinder a smaller share. This is especially required at full power. The result of the reduced receiver pressures is a sudden "drop" of pressure on exhaust to the receivers.

The diagram B is for full-power working; and it will be seen that the cut-off in the intermediate cylinder is considerably

later than in the previous diagram. The back-pressure line of the H.P. diagram is consequently lowered, and the area of that diagram much increased. The back-pressure line of the intermediate diagram is similarly lowered by a late cut-off in the low-pressure cylinder. There is considerable "drop" between the cylinders, and

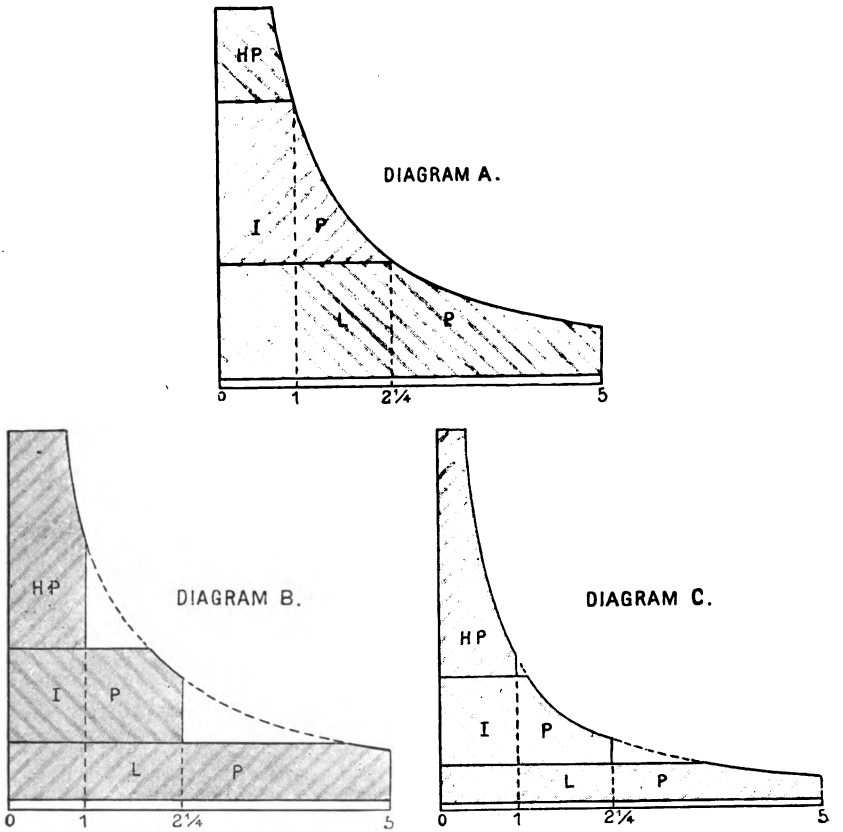


FIG. 89.

this means a loss of work corresponding to the unshaded areas, which loss is unavoidable if we are to divide the power fairly evenly.

At reduced powers the expansion is more nearly continuous. In reducing power by cutting off earlier in the H.P. cylinder, if other things were left unaltered, the work would gradually accumulate in that cylinder, the L.P. eventually doing very little; this we can prevent by cutting off earlier also in the later cylinders.

Diagram C shows the diagrams at low power, the cut-off in the H.P. cylinder being at $\frac{1}{2}$ stroke. The cut-off in the later cylinders has been advanced (though to a less extent in the low than in the intermediate) to raise the back-pressure lines, and so keep the work areas tolerably equal; and the drop between the cylinders is now very slight.

Thus in naval engines, at full power, cut-off is late in all cylinders, and there is considerable drop between the cylinders. In reducing power the cut-off is advanced in all cylinders; greater expansion is thus given, and the drop becomes less. The alterations required are not great, and they can be made fairly well by linking-up, especially where the link-adjusting gear described in Chapter VI. is fitted. An equal division of power, however, cannot usually be secured throughout. In general, notwithstanding the adjustments, the high-pressure cylinder does less than its share at the highest powers, and rather more than its share at the lowest powers; and the low-pressure cylinder the reverse.

Coal Consumption.—Notwithstanding the defect in the expansion previously described as existing in engines working on the compound plan, the system leads in practice to economy of fuel by making large expansion consistent with fairly efficient conditions of working in the cylinders; and the further the system is carried with suitable pressures, the more economical is the result.

The sea-going consumption per hour for each I.H.P. developed in good examples of service engines, working at economical powers and with good coal, is ordinarily about 2 lbs. for triple, and $2\frac{1}{3}$ lbs. to $2\frac{1}{2}$ lbs. for the later double-expansion engines; and in the best examples, under the most favourable conditions, $1\frac{3}{4}$ lbs. for triple, and 2 lbs. to $2\frac{1}{4}$ lbs. for double-expansion, while in the latest single-expansion engines it was rarely less than $3\frac{1}{2}$ lbs. At the low powers ordinarily used the consumption per I.H.P. becomes greater in all types, but the comparison between them remains much the same—showing, on the whole, a saving for the triple-expansion of 15 to 20 per cent of the expenditure in the double-expansion, and of 45 to 50 per cent of the expenditure in the single-expansion engine, each type working with a suitable pressure.

Outside the service, the consumption in triple-expansion engines in ordinary running is about 1·6 lbs. per hour per I.H.P.,

and in double-expansion about 2 lbs. The superiority of these over service engines in respect of fuel consumption will be understood from what has gone before.

It must be understood that where the coal available is of inferior quality the figures given above may be considerably increased.

The rate of Coal Consumption per I.H.P., which is usually taken as the test of economical working, depends, of course, upon the performance of the boilers as well as of the engines; but the efficiency of the boiler has not much improved for a long time past, and the improvement in economy may be regarded principally as the result of increased efficiency in the use of steam in the engines.

Summing up: the present system secures fuel economy in conjunction with smoothness of working. Triple-expansion engines are superior in these respects to double-expansion, because they give a higher degree of expansion to the steam, while retaining a moderate expansion in each cylinder. Double-expansion engines are greatly superior to single-expansion for the same reason.

Special Fittings (Figs. 90 and 91).—The I.P. and L.P. cylinders, with their pistons, covers, and valve casings, are designed and tested with reference to the reduced pressures with which they are intended to work, and not for the full working pressure, which is intended for the H.P. cylinder only. The following fittings are therefore provided to protect the various parts of these castings from over-pressure.

1. *Reducing-Valves,* K, K, *for the* I.P. *and* L.P. *Cylinder Jackets,* for limiting the pressure maintained in the jacket to a point corresponding with the working pressure within the cylinder. The reducing-valve is between the jacket stop-valve, J, and the jacket; and, the stop-valve being open, the reducing-valve allows steam to pass to the jacket only when the pressure within it has fallen below a certain point, closing automatically to stop the supply as soon as this pressure is regained. The jacket pressure at which the valve shall close is regulated by the adjustment of a spring on the outside.

2. *Safety-Valves,* S, S, *for the same Jackets,* to prevent over-pressure in the event of the reducing-valves failing to act properly. These are loaded to about 80 lbs. and 30 lbs. respectively (or less

in recent examples), with 155 lbs. working pressure. They blow off into the engine-room, and so call attention.

3. *Safety-Valves,* V, V, *for the Receivers,* to prevent over-pressure on the parts with which the receivers communicate. These also are loaded as above; they usually blow off into the air on deck; when this is the case, small sentinel-valves are fitted to blow off into the engine-room, to call attention.

4. *Pressure-Gauges* are fitted to indicate the pressures in the jackets and the receivers; on these are plainly marked the pressure which should not be exceeded.

The starting-valves, P′, P, shown in the same figure are simple *pass-valves,* which enable steam to be admitted by hand to the receivers for the intermediate and low-pressure cylinders. In the arrangement illustrated, P admits steam on the steam side of the intermediate slide-valve, filling the first receiver, and P′ on the exhaust side for the second receiver and L.P. slide-valve.

Combined Diagrams.—The diagrams from a set of compound cylinders, as originally taken, are not to the same scale either of pressures or of volumes; weaker indicator springs are used in the case of the lower-pressure cylinders, in order that those figures may have reasonable height; and the lengths of the figures are usually equal, whereas the cylinder volumes represented by those lengths are very different. This is of no consequence so far as the calculation for the power is concerned. For this purpose each pair of diagrams is measured, as before described, according to its own scale of pressures, and the mean effective pressure found is used to calculate the power developed in that engine; the results are then added together for the total power.

Useful information, however, may be obtained by bringing the diagrams to the same scale and combining them, that is, placing them in their relative positions, as in Fig. 92, and as in the theoretical diagrams, Fig. 89, already referred to.

The process may be briefly explained. In Fig. 92 are diagrams from a set of triple-expansion engines, a single diagram being shown from each cylinder for simplicity. The atmospheric lines of these diagrams are first lengthened or shortened to bring them to the same relative proportion as the cylinder volumes. The pressures are then taken from the diagrams, and are set up at the respective points of the stroke, a common scale of pressure being used throughout. In each case the pressure used at any point of

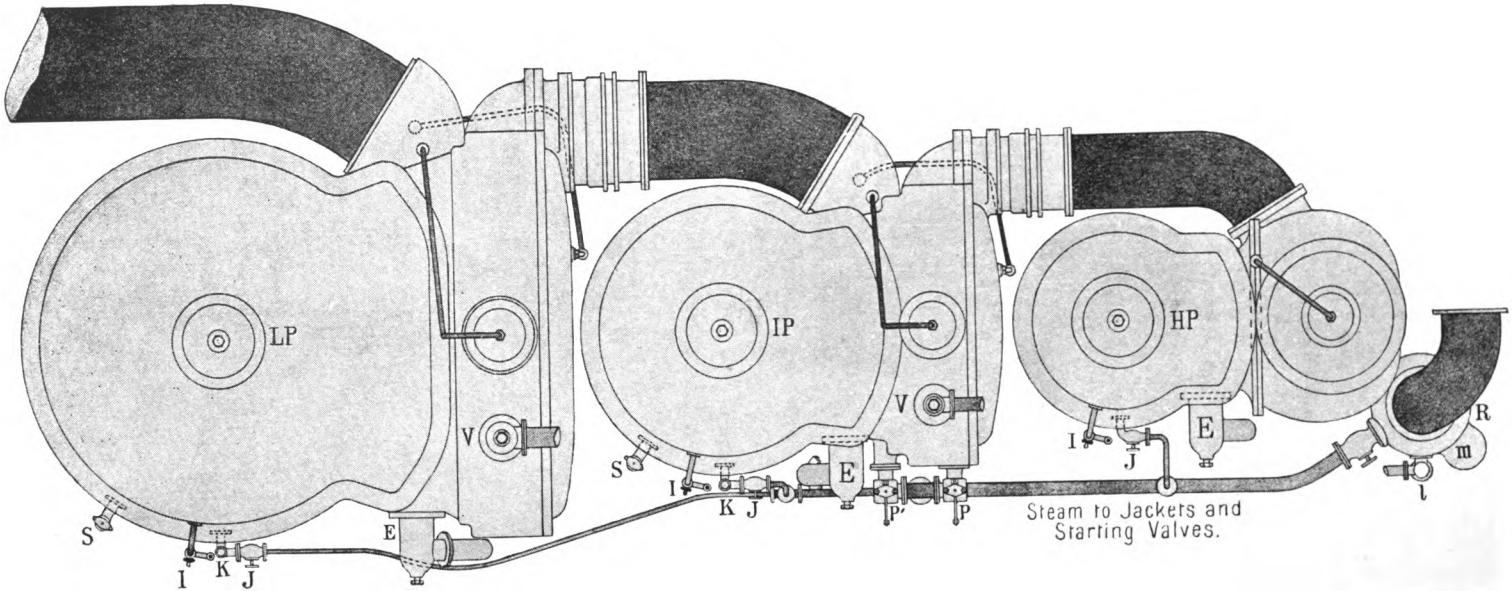

LP
IP
HP
V
V
S
S
I
K
J
E
E
I
K
J
P'
P
E
I
J
R
m
l
Steam to Jackets and
Starting Valves.

FIG. 90.

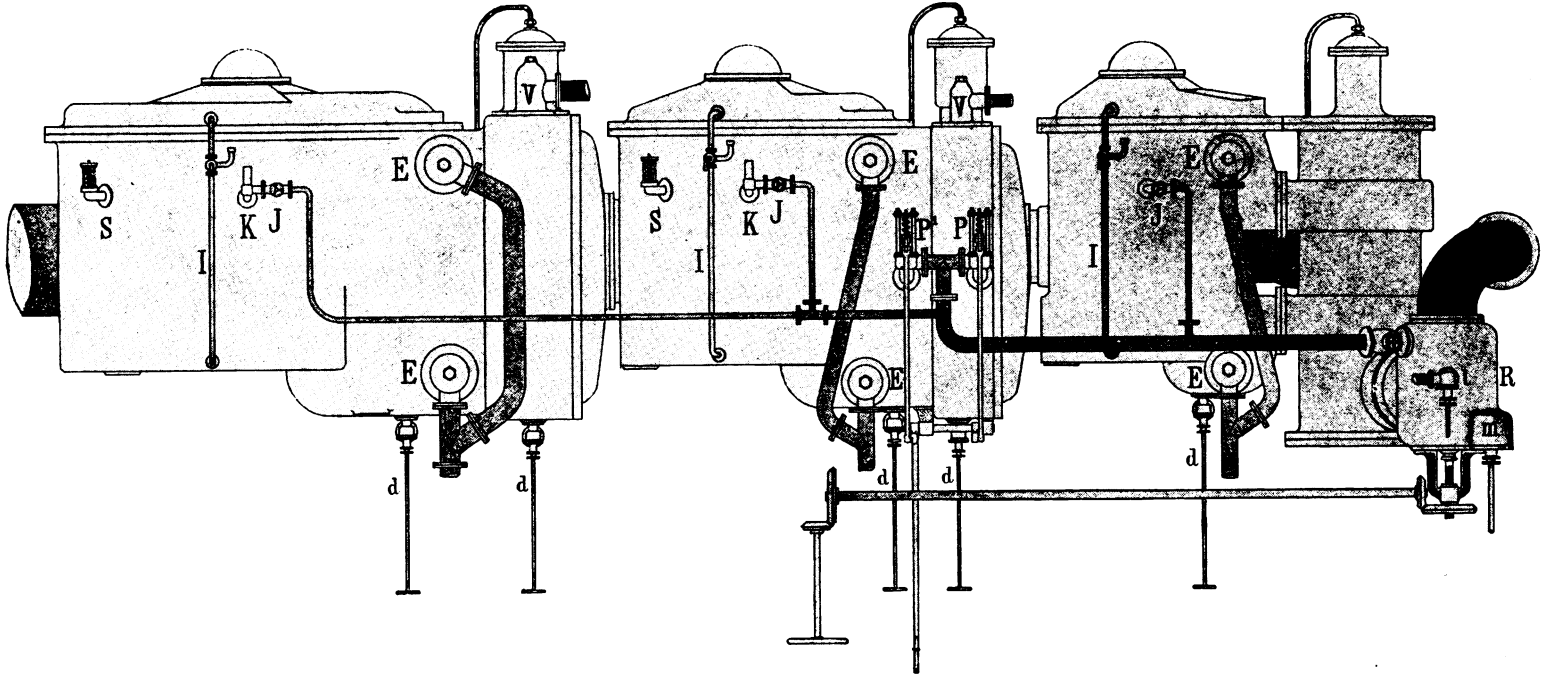


FIG. 91.

the stroke would be the mean for the two strokes. The new diagrams thus constructed are placed at their proper heights with reference to a common base line, OX. For accuracy they should also be placed at distances from a common vertical line, OY,

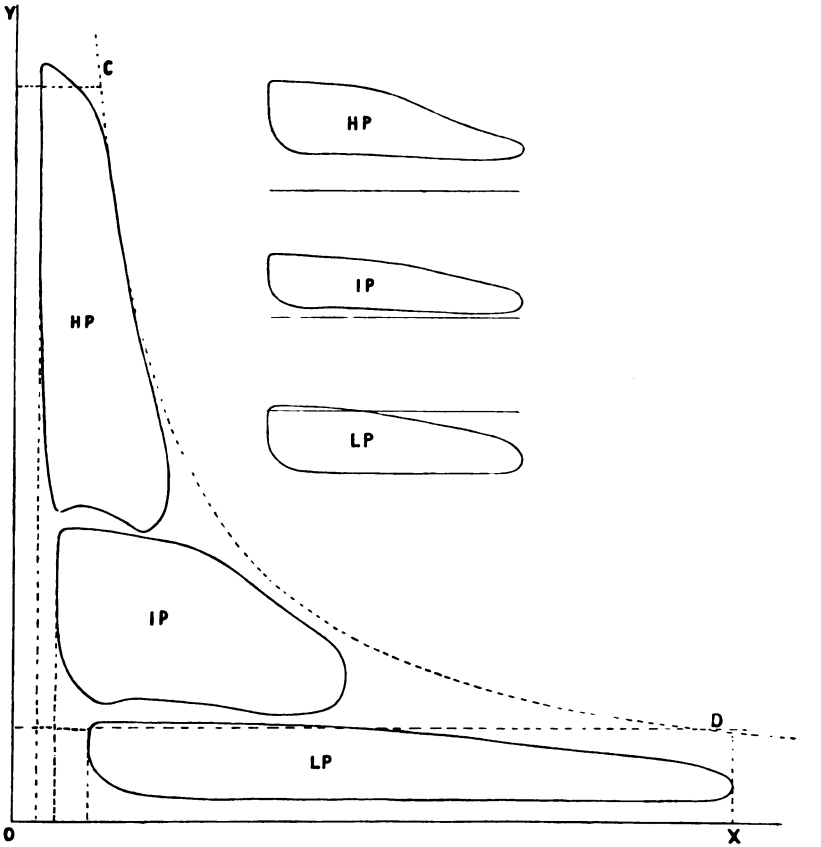


FIG. 92.

corresponding to the clearance volumes of the cylinders, since the steam supplied to the cylinders fills the clearance spaces also. The clearance usually amounts to about 22 per cent, 15 per cent, and 12 per cent in the H.P., I.P., and L.P. cylinders respectively, assuming a piston-valve on the high-pressure cylinder. For comparison, the dotted curve CD has been drawn to represent the expansion of the same steam according to the assumed law, pressure $\propto \frac{1}{\text{vol}}$.

The character of the actual expansion may thus be examined, and the loss of work area consequent on the gaps in the expansion and other causes becomes apparent. It is from the examination of such diagrams that engineers, in settling the dimensions for a new set of engines, are able to say beforehand what work may be expected per revolution with given dimensions of low-pressure cylinder, a given initial pressure, and a given ratio of expansion. If the theoretical expansion curve be continued up to the height corresponding with the boiler pressure, the loss of work due to fall of pressure from boilers to high-pressure cylinder will also be included.

CHAPTER X

CONDENSER AND FITTINGS

THE office of the condenser has already been stated. It may be briefly described as a large closed vessel with a supply of

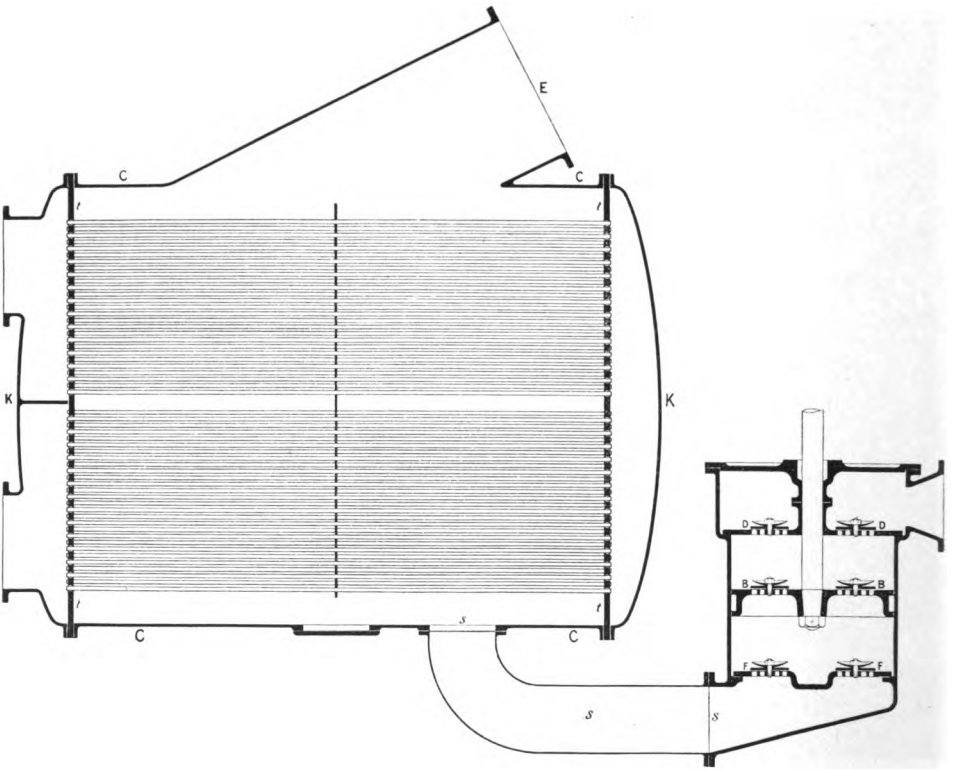


FIG. 93.

cooling-water constantly renewed, and with an air-pump to withdraw air and condensed steam from the interior.

In the *surface condenser*, which for the last thirty years has been in general use in ships, the steam exhausted to the condenser is kept separate from the sea water which is used for condensing it, in order that the condensed steam may be available as fresh feed-water for the boilers. The steam is condensed by contact

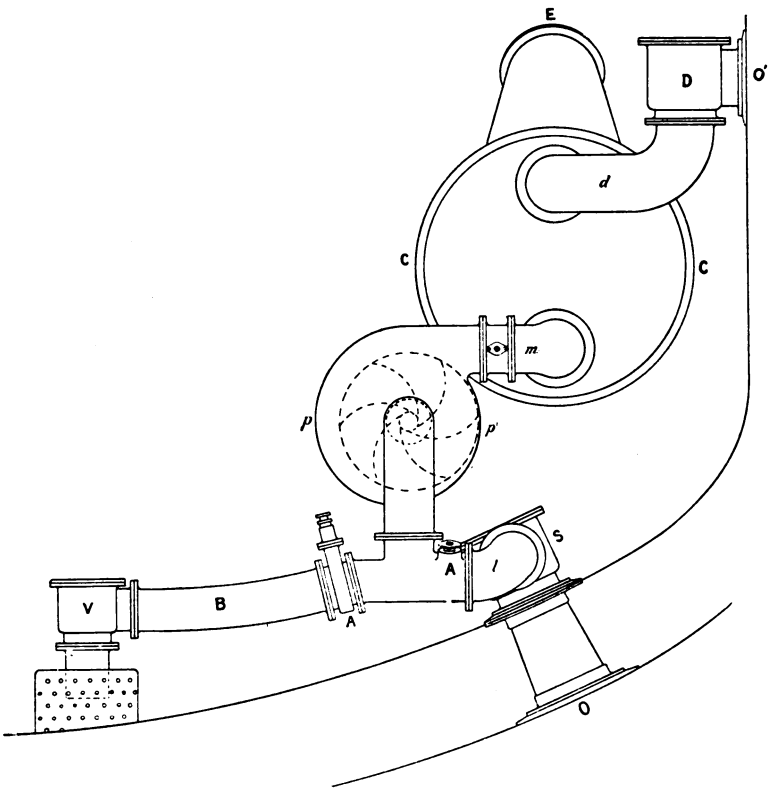


FIG. 94.

with the surface of a large number of thin brass tubes, kept cold by a continuous flow of cold sea water over the opposite surface of the metal. The tubes are contained within the *condenser casing*, CC, Figs. 93 and 94, to the upper part of which the exhaust-pipe is led. The figures show the usual arrangements in an elementary manner. The condensing-water is driven through the tubes, and the steam, entering at E, becomes condensed by contact with the outer surface of the tubes, and falls as fresh water to the

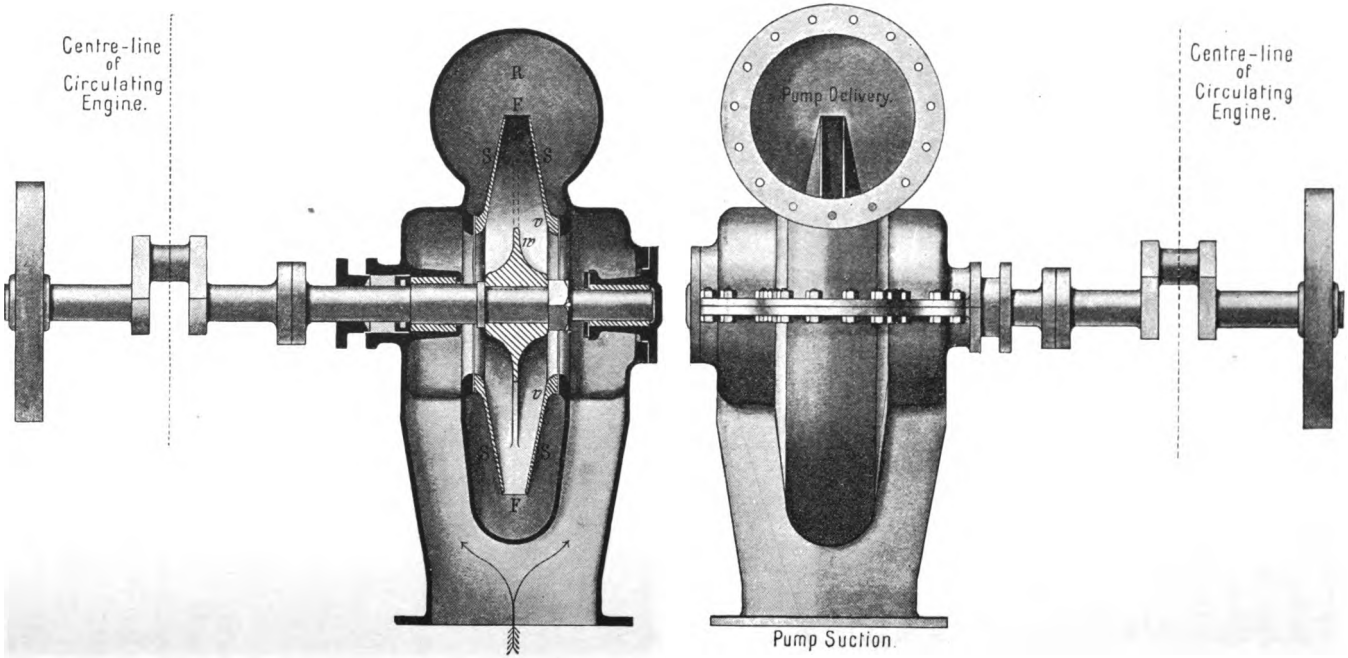

Centre-line
of
Circulating
Engine.
R
F
S
S
v
w
v
S
F
Pump Delivery.
Centre-line
of
Circulating
Engine.
Pump Suction.

FIG. 95.

bottom of the casing, whence it is withdrawn by the air-pump. Sometimes this arrangement is reversed, the water being on the outside and the steam on the inside of the tubes; but the plan shown is now the more general.

Circulating Arrangements.—The cooling-water is *circulated* (*i.e.* being received from the sea is delivered through the condenser again to the sea) by a pump, generally of the centrifugal kind. This pump is driven by an independent engine, an arrangement which enables it to be kept going while the main engines are standing under steam, and its speed to be easily regulated to suit the requirements at all times. Referring to Fig. 94, the water flows to the pump, *pp'*, by the inlet, O, and the suction-pipe, *l*, and is driven by the pump through its delivery-pipe, *m*, the condenser-tubes, the discharge-pipe, *d*, and outlet, O', overboard.

Inside the inlet and outlet orifices are valves opened by hand; S the inlet or main sea-suction valve, D the outlet or main discharge-valve. Both are usually ordinary screw-down stop-valves.

The centrifugal pump, Figs. 95 and 95A, consists of a fan or wheel, and a casing within which the wheel rotates; the whole is of gun-metal. The fan, FF, consists of a central web, *w*, on an ordinary boss, by which the wheel is mounted on the shaft; two side or cover plates, *ss*, slightly coned and annular; and arms or vanes, *vv*, which are cross-webs springing from the central web and boss, and extending to the side plates and to the circumference of the wheel. The vanes are curved as shown with reference to the direction of rotation. The parts mentioned are all in one casting, and frequently the shaft also is now cast with the wheel. Around the circumference of the wheel is a chamber or passage in the casing, R, of gradually increasing area, from which the delivery-pipe is led. The pump-suction communicates with the centre of

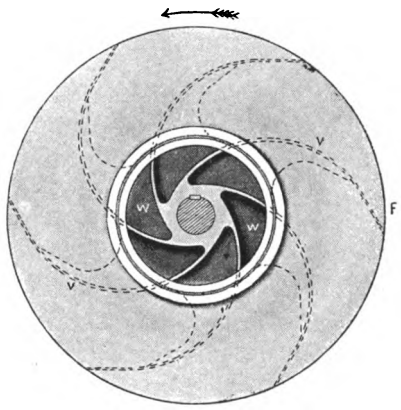


FIG. 95A.

the pump on each side; and the water, flowing to the centre and passing between the vanes, is rotated by them and delivered at the circumference of the fan to the passage, R, and delivery-pipe. This kind of pump works more smoothly and gives less trouble than an ordinary force-pump, from the fact that the motion is not reciprocating and that the pump has no valves; and it is very suitable where, as in this case, a large flow of water is required against a small head: for high lifts it would be unsuitable. In circulating the condensing-water there is no lift, and the only resistance is that due to friction of passages.

The centrifugal pump is intended, however, to be used also for pumping water overboard from the engine-room bilge in case of necessity. Here the lift is still moderate, *i.e.* from the bilge to the water-level. Large pumps like those used in first-class cruisers and battle-ships will discharge from the bilge at the rate of 1000 to 1200 tons an hour each, assuming that the passages remain clear. When merely circulating, they deliver about twice as much water, though running at only about half the number of revolutions: the difference is due to the absence of lift in the latter case.

In order to provide large bilge-pumping power, and to secure the advantage of duplication in the circulating arrangements, two centrifugal pumps and two engines are usually fitted to each condenser, each of the pumps with its engine being sufficiently powerful for circulating purposes. A suction-pipe (B, Fig. 94) from the bilge is fitted to each pump, and valves, A, are fitted for shutting the pump off from the sea and connecting it with the bilge, or *vice versa;* so that either or both of the pumps may be used for either purpose. The suction-valve, V, at the lower end of the bilge suction-pipe is non-return; so that water cannot pass to the bilge by way of this pipe.

Fig. 95 shows the arrangement of the pumps. The engines are inverted; this helps to keep the pumps available during an emergency, by raising the level of the cylinders. These are usually not less than 12 feet above the bilges in a large ship. The engines take steam by a branch from the main steam-pipe, and exhaust into the main-engine condenser, or, if this is not available, to the auxiliary condenser or atmosphere; they may also take steam from the auxiliary steam-pipe.

Condenser Detail.—The condenser is now entirely of brass in

service engines. This material being free from waste by corrosion the casing may be made thin, and therefore light, in accordance with the smallness of the pressure upon it, *i.e.* the external atmospheric pressure. Brass tubes also may be made very thin; and the material is a good conductor of heat.

The casing is usually of circular or oval section, and is either cast, or is built up of brass plate. At the ends of the casing are the tube-plates, *tt*, Fig. 93, and outside these are covers, *kk*, forming water-chambers into which the tube ends open. Doors in the covers, not shown, allow access for examination of the tubes; and in one of the covers are the orifices for the entry and exit of the circulating water. A web across this chamber separates the upper part of it from the lower; so that the water passes on by the lower tubes only and returns by the upper.

The tubes vary in length up to about 10 feet in large condensers, and are either $\frac{5}{8}''$ or $\frac{3}{4}''$ external diameter, and about $\frac{1}{20}''$ thick. They must be fitted water-tight in the tube-plates; and it is important also that they should be free to lengthen or shorten under changes of temperature. The plan of fitting used in the service is to form a stuffing-box at each end of the tube, as shown in Fig. 96. The hole through the plate is larger than the tube, except at the bottom, and the enlarged part is screwed to receive a small screwed ferrule, or gland, G, which is used to compress a ring of special packing placed in the space around the tube end. The tube ends so fitted usually remain tight for a long time. This is a matter of great importance; for, should leakage occur, the feed-water for the boilers becomes more or less salt.

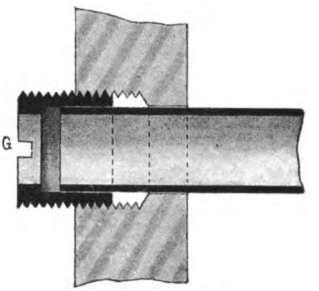


FIG. 96.

Air-Pump.—If a vacuum is to be maintained the condenser must be exhausted of air, not merely once for all, but continuously during the working of the engines; for air in greater or smaller quantity will find its way to the condenser with the steam, and, being non-condensable, will, if allowed, accumulate there. To withdraw air, and any other non-condensable gases present, is the primary office of the air-pump; and with these is withdrawn also the water resulting from condensation.

The pump is of the ordinary reciprocating kind, in which fluid is acted on by a piston moving along a barrel, with suction-valves opening into the pump and delivery-valves opening out from it for the passage of the fluid.

In all vertical engines the air-pump is worked vertically, and is of the *bucket* description, that is to say, in the piston are valves which allow the fluid to pass through it in one direction only, viz. from the suction to the delivery side, such a piston being technically called a bucket. In horizontal engines the pump motion is usually horizontal; and the pump is either of the solid piston (*i.e.* without valves in the piston) or of the plunger description.

The bucket-pump (Figs. 93 and 97) is superior to the others, and where a vertical motion can be conveniently given is invariably used. Its action is extremely simple: F are the suction or *foot* valves; D, the delivery or *head* valves; B, the *bucket* valves. All these open to allow a passage in one direction, viz. *outward from the condenser*, but close to prevent any return.

The bottom of the air-pump communicates with the bottom of the condenser by the suction-pipe and orifices, *s*, so that the water and air have free access to the under side of the foot-valves. During the *up-stroke* of the bucket, the foot-valves are opened by the slight excess of pressure on the condenser side of them, and water and air pass into the barrel below the bucket. At the same time, water and air above the bucket are driven through the delivery-valves; the bucket-valves being kept closed by the pressure above them. On the *down-stroke* occurring, the foot and delivery valves close, and the bucket-valves are opened by the pressure beneath them, thus allowing the bucket to pass to the bottom of the stroke through the imprisoned fluid. On the next up-stroke this fluid is delivered, while a fresh quantity flows into the barrel at the lower end, as before. The pump is therefore single acting, the delivery occurring only in the up-stroke.

The barrel and bucket are of gun-metal, as is the case in pumps generally. The valves require to be as light as possible, in order that they may lift readily under small pressures. They are either of vulcanised india-rubber, or, more generally now, of thin sheet metal, usually brass or phosphor bronze. Evidently the air-pump cannot reduce the pressure within the condenser below that which is sufficient to lift the foot-valves. The

vacuum attainable will therefore depend on the lightness of the valves and also on the thoroughness with which pressure is removed from above them; the latter point, again, depending on the air-tightness of the bucket, and on the *nearness with which the bucket approaches the foot and delivery valves at the ends of its stroke.* It

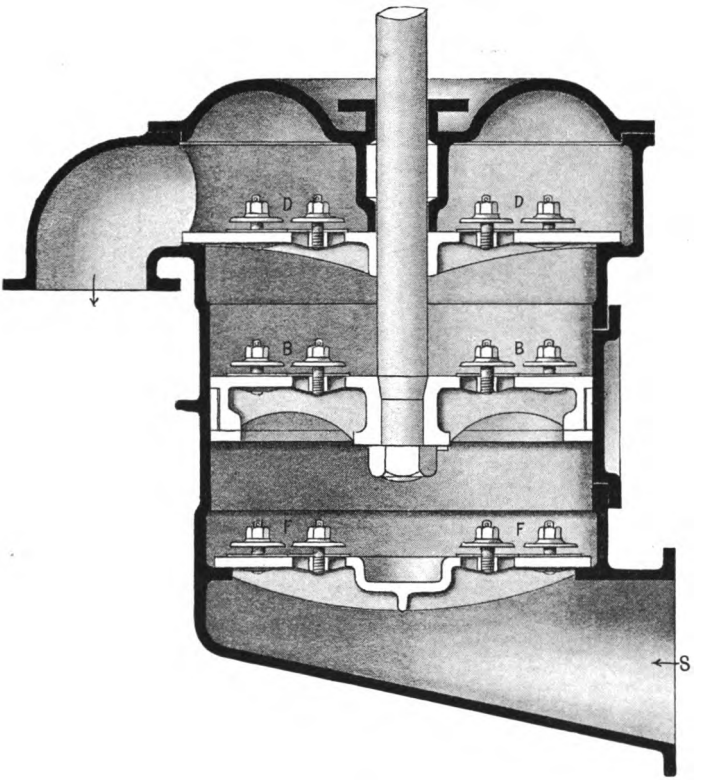


FIG. 97.

is in this last particular that the superiority of the bucket-pump lies.

The air-pump bucket is generally worked from the L.P. piston-rod head by means of a rocking-lever, which reduces the stroke by about one-half, and so keeps the speed of the bucket moderate. The arrangement may be seen in Fig. 33, where A is the air-pump, and L the rocking-lever. At the end of the latter are links, *l*, connecting it with the piston-rod head and with the air-pump-rod head respectively.

Feed-Tank.—The air-pump discharges into the feed-tank: a simple tank of galvanised steel plate, which acts as a reservoir for feed-water, being large enough to allow it to accumulate for a short time (say 3 to 5 minutes at full power, and proportionately longer at reduced powers), when not immediately required for the boilers. One is fitted in each engine-room. It is provided, as now fitted, with two overflow pipes: one leading to the reserve tanks for the preservation of the water, the other to the bilge for enabling oily matter accumulating on the surface to be got rid of.

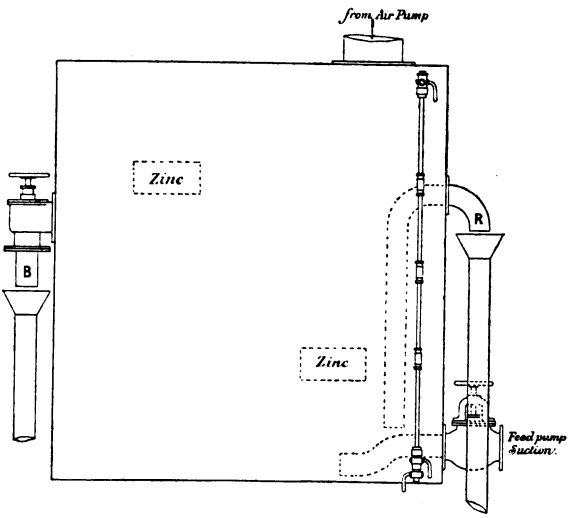


FIG. 98.

Other requirements are that there should be an escape for air, and that grease at the surface should not pass by overflow to the reserve tanks, nor to the feed-pumps for the boilers.

The sketch (Fig. 98) shows one arrangement of the fittings. B is the overflow-pipe to bilge, fitted with a stop-valve; R is the overflow-pipe to reserve tanks, having an internal pipe led to the bottom of the tank. The overflow to reserve tanks takes place from the bottom, and is inoperative until the water has risen to the upper part of the pipe, ceasing again when the water has fallen to the level of the open mouth R. The overflow to bilge may be brought into operation when desired. In other cases the bilge overflow is placed near the top of the tank, and is

without a stop-valve, which is fitted to the reserve tank overflow instead.

An open pipe, or some equivalent, is fitted at the top of the tank to permit the escape of air. The two tanks in a twin-screw ship are connected by a pipe, with shut-off valves worked from either engine-room. Gauge glasses are fitted to show the height of water in the tanks, placed so as to be visible from the starting-platforms.

Reserve Feed-Tanks, for carrying a reserve of fresh water, are provided by utilising the double bottom under the engine or boiler rooms. These are fitted with suitable connections for receiving water from the feed-tanks, from alongside (unless the supply from alongside is by way of the feed-tanks), and from the boilers; and also for the withdrawal of water when required. The quantity provided for is from 50 to 100 tons in large ships.

Feed-Pumps.—From the bottom of the feed-tanks the feed-water is taken by the feed-pumps as required, and delivered to the boilers.

Various changes have been made in regard to feed-pumps. Formerly the main pumps were worked by the main engines; now they are independent steam-pumps, worked by steam from the main steam-pipes, while the auxiliary pumps, also independent, take steam from the auxiliary steam-pipes. These independent pumps have commonly been placed in the boiler-rooms, one main and one auxiliary pump being fitted to each set of boilers, the main pumps delivering only to the set of boilers, and the auxiliary to all boilers. A more recent arrangement is to fit one main pump in each engine-room and one auxiliary pump in each boiler-room, each main pump delivering to all boilers, and each auxiliary pump only to a certain number of boilers.

The main feed-pumps have, at most, only one suction besides that from the feed-tanks, viz. one for withdrawing water directly from the condensers when the engines are standing (the air-pumps being then at rest) and the silent blow-off is being used; and this suction is now generally unnecessary, the water being withdrawn by other means (see Silent Blow-off, page 165). The auxiliary feed-pumps draw from the same places as the main pumps, and in addition are connected with the reserve tanks and with the sea.

The steam pumping-engines employed as feed-pumps have generally been of an ordinary description, similar to those used for

fire and bilge purposes. Pumps of special construction, however, after the pattern of the Worthington pump, have lately been largely used as feed-pumps. These are of large capacity, and work with few strokes per minute; they are also without a crank shaft. They not only work with greater smoothness and fill much better with water than ordinary pumps, running at the ordinary speeds, and are therefore less troublesome and more reliable, but their speed may also be easily regulated automatically. This is done by means of controlling gear connected with the steam regulating-valve of the pump, and worked by a float in the feed-tank. The gear is so arranged as to stop the pump when the water in the tank falls to a certain level, so that neither the oily matter on the surface, nor the air, may be delivered to the boilers.

Course of Water as Feed-Water from Condenser to Boilers, and as Steam through Engines to Condenser.—The entire circuit made by the working fluid in the steam-engine has now been considered; and it may be useful to give a brief summary, naming in order the principal parts through which it passes.

Condenser to Feed-Tank.	Air-pump suction-pipe. ,, ,, foot, bucket, and delivery valves. ,, ,, discharge-pipe. Feed-tank.
Feed-Tank to Boiler.	Feed-pump suction-pipe. ,, ,, suction and delivery valves. ,, ,, delivery-pipe, or feed-pipe, with branch to boiler; feed-valve. Internal feed-pipe.
Boiler to Engines.	Internal steam-pipe, boiler stop-valve. Main steam-pipe, bulkhead and screw-down stop-valves. Regulating-valve. H.P. slide-chest and slide-valve.
Through H.P. Cylinder.	H.P. steam-passage, cylinder, steam and exhaust passages. 1st receiver and I.P. slide-chest and slide-valve.
Through I.P. Cylinder.	I.P. steam-passage, cylinder, steam and exhaust passages. 2nd receiver and L.P. slide-chest and slide-valves.
Through L.P. Cylinder.	L.P. steam-passage, cylinder, steam and exhaust passages. Exhaust-pipe, condenser.

Here the steam is condensed by contact with the cold tubes, and, falling to the bottom of the condenser casing as fresh water, recommences the circuit.

CHAPTER XI

FEED-WATER FOR BOILERS—BOILER PRESERVATION

Feed-Water.—In high-pressure boilers the use of fresh water is practically a necessity. The reasons for avoiding the use of sea water will appear from what follows.

The salts dissolved in sea water, amounting to about $\frac{1}{32}$ by weight, though consisting largely of common salt (about $\frac{4}{5}$ of the entire amount), include also salts of lime and of magnesium, as well as some others. These salts are only of consequence when they appear as solid matter; as may occur from the concentration of the salts due to the evaporation of the water, or from high temperature alone. The common salt gives no trouble in boilers, for it is soluble at all temperatures in a very much larger proportion than that existing in sea water; it therefore remains in solution under all ordinary circumstances.

The most troublesome ingredient is sulphate of lime, which is present only to the extent of about $1\frac{1}{2}$ parts in 1000, or say $3\frac{1}{2}$ lbs. per ton of sea water. This salt is readily thrown out of solution, the precipitate forming a hard, badly-conducting scale on the internal surfaces, which, at the best, obstructs the passage of heat to the water, and is further liable to cause over-heating of the plates. The solubility of sulphate of lime in water, slight even at low temperatures, diminishes as the temperature is raised, until, at about 290° Fahr., the salt becomes entirely insoluble. The magnesium salts similarly become insoluble at high temperatures; but the precipitate, though objectionable in other ways, does not readily form scale.

At the temperatures formerly common in boilers, when the working pressure was only 15 to 20 lbs. by gauge, the water could hold in solution, say, one-and-a-quarter to one-and-a-half

times the quantity of lime-sulphate naturally present in sea water. Sea water could therefore be used as feed, and concentration of the salts by evaporation could be permitted to a slight extent; for so long as the proportion of salts present did not exceed one-and-a-half to two thirty-seconds, the formation of scale would take place only slowly. This degree of saltness would be shown by the service hydrometer as a density of 15 to 20; the number 10 representing the natural density of sea water, and zero that of fresh water, at the temperature of 200° Fahr. in each case.

In such boilers, fed with salt water, the density was kept down by the systematic "blowing-out" of a portion of the super-salted water into the sea,—a process entailing a waste of heat, which amounted often to 15 or 20 per cent of the whole heat transmitted to the water.

In high-pressure boilers the temperature of the water is, say, 330° to 380° according to pressure; and, as a consequence of the temperature, *any sulphate of lime admitted* with the feed-water will be deposited, and scale will be formed to a corresponding extent, while the effect of even a slight scale is much more serious than was the case formerly. It is therefore important that the introduction of sea water, even in the smallest quantities, should be avoided if possible.

Some figures on this point may be useful.

In a modern ship of say 12,000 I.H.P., the amount of feed-water passed into the boilers for all purposes would probably be in round numbers as follows:—

At 10,000 I.H.P. about 2000 tons per day
„ 6000 „ „ 1000 „ „
„ 3000 „ „ 630 „ „
or 1 ton per day for 5 to 6 I.H.P. continuously developed.

Assuming that, by slight leakage at the condenser tube ends, the density of the feed is raised only from 0 to 1 by hydrometer (*i.e.* to one-tenth the saltness of sea water), the amount of lime-sulphate passed into the boilers, reckoned at $\frac{1}{3}$ lb. per ton of feed-water, will amount to 660 lbs., 330 lbs., and 210 lbs. per day respectively, at the above powers, while the magnesium salts, which will also appear as solid matter, will be very much greater still in quantity.

In view of the above facts, ships' boilers are now supplied with fresh water, surface condensers are fitted to preserve the

freshness of the feed, various arrangements are used for minimising waste of fresh water, and the unavoidable waste is made up by fresh water from the distilling apparatus, or from the reserve tanks.

Arrangements for Minimising Waste of Fresh Water

1. *Auxiliary Condensers.*—These are provided to condense and preserve for the boilers the steam used for the various auxiliary purposes in the ship, which steam was formerly exhausted to the atmosphere. Two condensers are fitted, one in each engine-room. These are precisely like the main condensers, both in construction and action, and each has its own circulating and air-pumps, driven by an independent engine. The water produced by condensation is delivered by the auxiliary air-pump to the feed-tanks for the boilers.

The auxiliary steam-engines are now so numerous in war-ships that the condensing surface required in these condensers is as much as 10 to 12 per cent of that necessary in the main condensers, although the auxiliary engines connected with the working of the main engines—such as centrifugal-pump engines, starting engines, and main-feed engines as now fitted—are provided for by the main condensers, and, of the others, only a certain number would be at work at any one time.

2. *Silent Blow-off.*—This fitting has already been described. When the engines are suddenly eased or stopped, and the steam cannot otherwise be kept from blowing off at the safety-valves, the excess steam may be blown into the main condensers and there condensed. The water produced cannot pass from the condenser in the ordinary way; but a suction to the *auxiliary* air-pump, which is worked by an independent engine, enables that pump to withdraw the water and deliver it to the feed-tank; in other cases it is withdrawn direct by the feed-pump.

3. *Feed-Tanks as Reservoirs, with Overflow to Reserve Tanks.*—By affording space for the feed-water to accumulate in, these tanks prevent the frequent discharge of water to the bilge, or overboard, which took place from the comparatively small hot-wells formerly in use.

4. *Preservation of Drainage Water.*—The drains are connected either directly with the main or auxiliary condensers, as

in the case of the principal main-engine drains, or with a common drain-pipe leading to a drain-tank, as in the case of the numerous steam and exhaust pipes connected with the auxiliary engines. From the drain-tank the water is returned to the feed-tank by a small pump. The drainage water was formerly largely wasted; that from the *steam-cylinders* of auxiliary engines is still discharged, being too greasy to be returned to the feed-tank.

5. Blowing out water from the boilers at work is no longer necessary, except to a slight extent when the boilers have been at work for some time, and the water is becoming too dirty, principally from the accumulation of oily matter. The surface blow-out is then used, but not more than is absolutely necessary.

Notwithstanding the above arrangements, a very considerable waste of water does occur, even where the machinery is in good order and great care is exercised. In ordinary cases not less than 4 to 6 tons per day per 1000 I.H.P. are required to make up the deficiency. This will amount, say in a 12,000 I.H.P. ship working at 4000 I.H.P., to no less than 16 to 24 tons a day. It must be remembered, however, that the amount of water then being used per day for all purposes is probably 800 tons; so that the waste represents 2 to 3 per cent of the water used. This must be accounted for by small leakages of steam and water here and there at the numerous glands, joints, escape-valves, and drains throughout the ship. The drainage at the cylinders of the auxiliary engines is of necessity wasted. The waste, whether small or great, must be made up by fresh water, preferably distilled water, if scale is to be avoided.

Distilling Arrangements.—Formerly fresh water was made for the ship's use by evaporating sea water in one of the ship's boilers, and condensing the steam in a Normandy distilling condenser. This practice was bad for the boilers employed, inasmuch as the deposition of the salts took place within them. To avoid this the plan of *double distillation* was introduced, which is as follows:—

Evaporation of sea water is produced by means of *steam*, which is admitted to tubes situated within a special evaporating vessel. The sea water surrounds the tubes, and is not circulated through the vessel, but merely fills it up to a certain level; heat is transmitted from the steam, and evaporation slowly takes place outside

the tubes at a lower pressure than that of the steam within them. The new or *secondary* steam thus formed is passed into the condensing part of the apparatus, producing *gained* fresh water; while a gradual condensation of the original or *primary* steam takes place within the tubes, the supply to which is steadily maintained. The condensed steam passes to the auxiliary condenser for the feed-tanks, and so is returned to the boilers.

The important feature is that here the evaporation of the sea water is carried out in a vessel in which there are no critical parts, such as the furnaces of a boiler, to be injured by scale, and which is so constructed that it may be easily opened and cleaned. Several forms of double-distilling apparatus, consisting of an evaporating vessel and a condensing vessel, are in use. At the present time Weir's Evaporator, used in conjunction with the ordinary Normandy Condenser, is the most commonly fitted in ships in the service; a more detailed description of this evaporator may therefore be desirable.

Weir's Evaporator (Fig. 99).—This vessel consists of a cylindrical steel shell, *ss*, like a boiler shell, within which, at the lower part, are situated copper tubes, *tt*, bent into a U shape, as

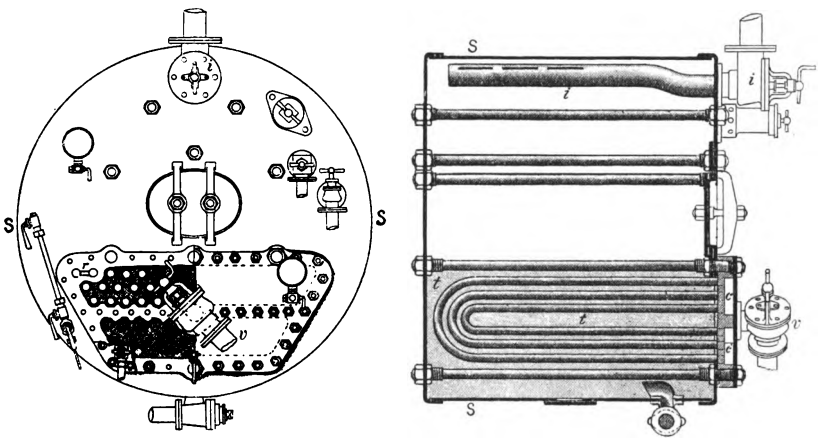


FIG. 99.

shown. The ends of the tubes open into the two chambers, *c*, *c'*, of a casing carried by the tube-plate, and the latter is jointed to the front of the shell. To one of the chambers, *c*, steam is supplied for the tubes by the pipe and regulating stop-valve, *v*; while from

the other, *c'*, the condensed steam is drained away; being passed, however, once more through the water by means of a *return*-tube, *r*. The exit ends of the tubes are nearly closed by plugs, only a very small hole being left in each plug as an outlet.

The tubes are kept covered by sea water, supplied by a pump and regulated by a feed-valve; and the secondary steam produced is withdrawn by means of the internal pipe and stop-valve, *ii*, and is passed to the Normandy distilling condenser for producing drinking-water, or to the auxiliary condenser for producing "make-up" feed for the boilers, while the condensed primary steam is also returned to the boilers. This evaporator is supplied with all the ordinary mountings of a boiler, as shown in Fig. 99. During working, the density is kept down to 30 or less by brining; and, when necessary for cleaning, the nest of tubes may be entirely withdrawn by breaking the joint between the tube-plate and the front-plate. The inside of the shell is then also easily accessible.

The Normandy Condenser consists of a casing containing vertical tubes to the inside of which the steam to be condensed is admitted, while cold sea water is circulated through the casing outside the tubes by means of a pump.

The entire distilling apparatus is fitted in duplicate, and a spare set of tubes is carried for evaporator and condenser.

The amount of water actually gained in the double-distilling apparatus in ordinary working is about 5 lbs. to $5\frac{1}{2}$ lbs. for $7\frac{1}{2}$ lbs. to 8 lbs. of primary steam used, that is, for a pound of coal burnt. By connecting the two evaporators so that the secondary steam produced in one of them passes to the other as primary steam, producing there a further quantity of secondary steam, this result may be considerably increased.

The total water-producing capacity of the two evaporators is about $1\frac{3}{4}$ to 2 tons per hour in battle-ships (increased in recent cases), $1\frac{1}{2}$ to $1\frac{3}{4}$ tons in first-class cruisers, and 1 to $1\frac{1}{4}$ tons in second-class cruisers. These quantities, however, assume that both evaporators are available and efficient, which will probably not be the case at any given time, seeing that each requires scaling after a few days' work, and falls off in efficiency as the scale accumulates. The total capacity of the distilling condensers is not more than half the above quantities.

BOILER PRESERVATION

Brief Statement of Main Facts.—When boilers are in use, internal corrosion may be produced by chemical action or by galvanic action; injury may also be caused by the mechanical strains resulting from rapid and frequent changes of temperature.

Chemical Action.—This may arise from an acid condition of the boiler-water, or from the presence of air. The acidity may be caused by the introduction of sea water to the boilers, or by lubricants used in the cylinders. The treatment is, to use only mineral oil for cylinder lubrication, to exclude sea water from the boilers as completely as possible, and to keep the boiler-water in an alkaline condition by the addition of soda carbonate or lime to the water, if this should be found necessary.

The introduction of air into the boilers has been much reduced by the use of feed-tanks with provision for the escape of air, by using independent feed-pumps (which, being worked only when there is water in the tanks, are not so likely to draw air as were those formerly in use, worked by the main engines), and by avoiding blowing-out and other sources of waste of water as much as possible, so as to retain the boiler-water with the minimum of change.

Galvanic Action.—The most effectual means of preventing corrosion consists in the use of zinc, slabs of which are suspended within the boiler, especially in the water spaces, and placed in metallic connection with the internal surfaces, so as to form a galvanic couple with the plates. The use of zinc was adopted as a protection against galvanic action. It is certain that its effect is most beneficial, though to what extent the corrosive action prevented by it is of a galvanic or of a chemical nature is uncertain. The protection it affords arises from the plates being placed by connection with the zinc in an electro-negative condition. The zinc becomes wasted, and is renewed on ordinary service at intervals of about three to six months, depending on the amount of steaming.

Injury from Changes of Temperature.—To minimise the wear and tear from this cause, the raising of steam in the boilers rapidly, emptying the boilers or drawing their fires after steaming while they are still hot, and frequent changes of the

boilers in use, should all, as far as possible, be avoided. Under ordinary circumstances the boilers are closed up when done with, and left to cool down thoroughly before being emptied (if they are to be emptied) of water; the drawing of the fires and sweeping of the tubes being also delayed until the boilers are comparatively cool. The water in those boilers which are to be emptied, if clean enough to be preserved, is run into the reserve tanks.

If, on any occasion, forced draught must be used, it should be with great care and moderation. The presence of any deposit on the heating surfaces greatly increases the risk involved in using forced draught. As little oil as possible is now used for cylinder lubrication; for it is found that the oil produces in the boilers greasy deposits of a highly nonconducting character, which attach themselves to the heating surfaces, and are liable to produce serious overheating.

Boilers out of Use suffer most from exposure to air while in a damp condition. They are in general kept quite full of water, unless they are likely to remain out of use for a considerable time, in which case they are closed up as tightly as possible, after being thoroughly dried. Trays of burning coke are placed inside, on closing up the boiler, in order to consume the oxygen.

On the whole, the measures indicated above, which have been adopted as the result of experience and of the investigations of two Admiralty Boiler Committees, have proved efficacious in preventing the rapid and severe corrosion which was at one time so serious a matter, especially in naval boilers.

CHAPTER XII

COMBUSTION AND FUEL — EFFICIENCY

IN its ordinary practical sense the term combustion means a rapid and vigorous chemical union of combustible substances with *oxygen*, attended by the development of large quantities of heat. The necessary conditions for combustion are that the combustible and the oxygen should be brought into contact and raised to a sufficiently high temperature.

The combustible constituents of the fuels wood, coal, and oil, are the elements carbon and hydrogen. Some particulars of the combustion of these elements will first be desirable.

Hydrogen combines with oxygen in the proportion 1 to 8 by weight, producing water vapour, and developing about 62,000 B.T.U. per lb. of hydrogen.

Carbon combines with oxygen either in the proportion 12 to 32 by weight, or, if the oxygen present be insufficient for this, in the proportion 12 to 16. In the first case the combustion of the carbon is complete, and the result is the formation of carbonic-acid gas and the development of 14,500 B.T.U. per lb. carbon. In the second case the combustion of the carbon is incomplete (an atom of carbon having combined with only one atom of oxygen, instead of with two); the result is *carbonic-oxide* gas, and only 4400 B.T.U. per lb. carbon. The combustion of the carbon may, however, in this case be completed by supplying the carbonic-oxide gas with oxygen while at a proper temperature; in which case the result will be carbonic-acid gas as before, and the development of the remainder of the 14,500, B.T.U. per lb. carbon due to complete combustion.

The combustion of carbon may thus be completely effected in one or in two stages, with the same final result; but if in-

complete combustion be not completed there is a large sacrifice of available heat.

Coal consists largely of carbon, part of which is uncombined, while the other part is in combination with hydrogen, forming with it various hydro-carbon compounds. There are also impurities, the most important of which are the earthy matters which produce ash and clinker. The quantity of hydro-carbon matter may vary in extreme varieties from almost nothing up to half the entire coal.

The principal facts bearing upon the combustion of the hydro-carbon constituents of coal are as follows:—

1. Though solid at ordinary temperatures, the hydro-carbons are volatile, taking the form of gas when heated to temperatures which are, in most cases, well below that necessary for combustion.

2. The gases consist of various compounds of hydrogen and carbon, some richer in carbon than others. Most of these are very unstable, and decompose readily under the action of heat at temperatures which are common in boiler furnaces; the effect being to separate out more and more of the carbon from the compounds as the temperature is raised, and thus to produce free carbon and compounds less rich in carbon than before. If exposed to still higher temperatures the hydrogen and carbon become entirely separated.

3. Both hydrogen and carbon have a stronger affinity for oxygen than for each other; so that if the hydro-carbon gases be raised to the temperature of ignition in the presence of sufficient air, the hydrogen and the carbon, leaving each other, will combine separately with oxygen; in other words, will burn. If, however, the supply of oxygen be insufficient, some of the hydro-carbon matter will escape unburnt, but in a more or less decomposed condition, the carbon having been separated out to a greater or less extent according to the temperature to which the gases have been exposed.[1] In this case smoke is produced (see Smoke Production, page 173).

Hydro-carbon matter can only be burnt after taking the gaseous form. Coal gas, as supplied for lighting purposes, consists of the lighter of the hydro-carbon compounds in a gaseous

[1] It is generally stated that, with a limited air-supply, the hydrogen burns in preference to the carbon, being the more ready and vigorous combustible; according to recent research this appears doubtful. See Smithells on "Structure of Luminous Flames," *Journ. Chem. Soc.*, vol. lxi.

form, mixed with hydrogen gas; while the denser compounds are represented by coal-tar, from which the pitch, naphtha, and other products of the gas-works are obtained.

Combustion of Coal in Furnaces

1. *Hydro-Carbons: Smoke Production.*—When coal is thrown on the mass of glowing fuel already in the furnace, the heat of the fire liberates the hydro-carbons in the gaseous form, and raises the temperature of the gases. Assuming a fairly bituminous coal for illustration, the gases come off at first in large volumes, and afterwards more slowly, until the coal is practically free from volatile matter. If air be present in proper proportion, and be thoroughly mixed with the gases, as the latter reach the temperature of ignition the hydrogen and the carbon combine separately with oxygen, producing respectively water-vapour and carbonic-acid gas. In this case there is complete combustion of this part of the coal, and a complete absence of smoke. If, however, the supply of air be insufficient, or if the air do not penetrate and thoroughly mix with the gases, more or less of the carbon separated out by the heat of the fire will pass away unburnt, the result being not only a loss of fuel, but also the production of smoke. The separated but unburnt carbon from the hydro-carbon gases is in an extremely finely divided but solid state; and this carbon, blackening the cloud of vapour and gas issuing from the funnel, constitutes smoke. Deposited in the tubes and uptakes, it appears as soot.

Smoke can be prevented only by supplying air in sufficient quantity to the gases while they are sufficiently hot to burn. This, however, is not easy; the gases pass away very quickly from the furnaces and combustion-chambers, where the air should be supplied; and, for completeness of combustion, the air must penetrate and dilute the gases so that oxygen may come in contact with every part of them. For this purpose it is found necessary, in practice, to admit considerably more air than the amount chemically sufficient for the combustion of the elements; and, even so, in ordinary working with a smoky coal, it is almost impossible to entirely prevent smoke.

The air for the combustion of the gases must generally, in order to have sufficiently free access to them, be admitted directly into the

furnace, above the grate. This is the principal reason for the air passages through the furnace door. Plans have been tried by which the air-openings may be varied to suit the requirements of the furnace in different stages of the combustion; but such arrangements require closer attention than can generally be given to them in a stokehold. In practical working, usually, a number of holes through the plates of the door provide a constant passage.

If *too much* air be admitted above the grate, as, for instance, while the furnace door is open, the gases may be cooled down below the temperature of ignition; and in this way waste, and more or less smoke, may be caused, apart from the injurious effects on the furnace plates.

2. *Uncombined Carbon.*—This part remains as a mass of solid fuel on the grate; until, as oxygen comes into contact with the heated surfaces, the carbon particles burn, passing gradually away in the carbonic-acid gas formed.

The air for the combustion of this solid matter is supplied at the bottom of the fire, and the carbon with which it comes into contact enters into full combination. It is usually not difficult to secure the full combustion of this part of the coal; but with heavy fires, and with an insufficient draught, the carbonic-acid gas formed at the lower part of the fire may become reduced to carbonic-oxide by taking up a proportion of the hot carbon lying above, with which it comes in contact. Thus the final effect may be incompleteness of combustion of the carbon, and a large waste of heat, unless air be supplied to the carbonic-oxide gas to complete the combustion.

The direct air-supply above the grate is thus sometimes useful for completing the combustion of the carbon in the later stage, as well as for the combustion of the gases in the earlier stage of the process of coal combustion.

Varieties of Coal.—These range from highly bituminous coals at one end of the scale to non-bituminous, or anthracite, at the other end. The most notable and important feature in passing from the one extreme variety to the other is a gradual diminution in the proportion of hydro-carbons present, and an increase in the proportion of free carbon, the hydrogen in the less bituminous varieties being both less in quantity and combined with a smaller proportion of carbon.

The medium descriptions are alone suitable for use in steam-

boilers, the highly bituminous coals containing too much hydro-carbon matter, which makes them more suitable for gas-making, while the anthracite contains too little. The British steam coals may be divided, broadly, into two classes: *Newcastle or North Country* and *Welsh.*

1. *North Country Steam Coals.*—These are semi-bituminous coals containing about 30 per cent of hydro-carbon matter, and about 55 per cent of free carbon. The coals must therefore be largely burnt in the form of gas. For this reason it is difficult in practice to secure their complete combustion and to prevent smoke. The boiler-tubes also rapidly become fouled by soot. Many of these coals cake or bind together on the bars, and so impede the passage of air; this entails great trouble in breaking up the mass of fuel, and also necessitates frequent and prolonged opening of the furnace door, which is a source both of loss and of injury to the boiler. The presence of the bituminous matter, however, causes these coals to ignite easily and burn vigorously, many kinds burning with considerable flame.

2. *Welsh Steam Coals.*—These are slightly bituminous coals, sometimes called Anthracitic. They contain 10 to 15 per cent hydro-carbons and 75 to 80 per cent free carbon. It is therefore less difficult to secure their complete combustion and prevent smoke than is the case with the North Country coals. The deposition of soot is also less, and the temperature produced is higher. As a consequence, a higher evaporative result is generally obtained from them in practice. Under good circumstances afloat, about 9 lbs. of water may be evaporated per lb. of good Welsh coal, and about 8 lbs. of water per lb. of good North Country.

The evaporative superiority of Welsh coal for a given bunker space is still greater; for the coal is denser, so that a greater weight may be stowed in a given space, the difference being about 7 per cent. The Admiralty figures are 40 and 43 cubic feet per ton.

The comparative absence of bituminous matter, however, causes the coal to ignite less readily and burn more slowly than is sometimes desirable, especially if it is not fresh. This reduces the superiority of the Welsh coal so far as steam generation per hour from a given grate is concerned; and if the coal is not fairly fresh, it may be inferior in this respect to North Country coal; for it suffers greater deterioration from knocking about and

exposure, becoming more largely reduced to dust, and losing to a larger degree its original good qualities. Welsh coal does not, as a rule, cake to any objectionable extent.

On the whole, having regard to the principal requirements for marine purposes, good Welsh steam coal is to be preferred to any other kind. Of the British coals, Welsh steam coal only is at present supplied to naval vessels, both at home and abroad; but a knowledge of the characteristics of bituminous coal is of importance, since much of the native coal available on foreign stations is of that character, though generally inferior to our North Country steam coal. Some of the New Zealand coal, however, is equal to the good descriptions, and some of the Australian coal is little inferior to ordinary descriptions of North Country steam coal; these coals are largely used on the Australian station.

Anthracite.—This coal contains about 90 per cent free carbon, and is almost entirely free from hydro-carbons; it is consequently smokeless, but is difficult to ignite, and burns with extreme slowness; it also produces an extremely high temperature locally, which is injurious to the furnaces and fire-bars.

Patent Fuel.—The best kinds consist of the small of good Welsh coal compressed into the form of a brick, and they correspond very closely in their behaviour in the furnace, and evaporative results, with Welsh steam coal. Inferior kinds produce much smoke.

Liquid Fuel.—A few remarks with reference to liquid fuel may be of interest. Mineral oil, consisting entirely of hydro-carbon compounds in a liquid state, may, by suitable arrangements, be used as fuel in steam-boilers. It has been so used in Russia for many years, both in locomotives and in steam-vessels on the Caspian and Black Seas, the oil being there obtainable in great quantities and at a low price. The oil used is the residue left after the more volatile and dangerous parts of the crude petroleum have been distilled off. The usual arrangement is as follows:—The oil is "sprayed" into the furnace by a steam jet, and is accompanied by the air necessary for combustion, the flow of which is induced by the jet. An ordinary fire is at first kindled on the grate to start the furnace and produce steam of sufficient pressure for the jet, after which oil only is supplied as fuel. The oil is vaporised and burnt in the manner described for the volatile part of coal; and as the admixture and regulation of the air presents

no difficulty, perfect combustion may be easily secured. A ton of the oil is equal in evaporative effect to fully $1\frac{1}{2}$ tons of coal, and it occupies less space per ton in the ship.

Other obvious advantages of a liquid fuel for ships are: the ease with which it can be put on board, even at sea; the great simplification in the system of supplying fuel to the furnaces, and the consequent saving in labour, as well as the regularity and easy control of the supply; the absence of smoke, and of the necessity for cleaning fires.

On the other hand, apart from the fact that there is at present no large supply of mineral oil available in this country, or in the Colonies, the difficulty of carrying the oil with safety, and the added danger from its presence in action, would doubtless constitute serious objections to its use.

In this country, at present, liquid fuel cannot compete in respect of price with coal, though in some special cases where it exists as a waste product it is utilised as fuel. It is sometimes used in conjunction with an ordinary coal fire.

Value of Fuel—Proportion Utilised

A. A pound of good coal gives, by complete combustion, about 14,500 B.T.U. The *full equivalent* of this in work is $14{,}500 \times 772$ ft.-lbs.; and if we spread this performance over an hour, and convert to the corresponding horse-power, we find that *each pound of coal burnt per hour is worth rather over* $5\frac{1}{2}$ I.H.P. continuously developed over that period $\left[\frac{14{,}500 \times 772}{60 \times 33{,}000} = 5{\cdot}65\right]$, if the whole heat-energy due to the fuel could be utilised.

This gives a standard with which practical results may be compared. In a modern naval vessel, under good conditions, one horse-power may be produced by an expenditure of say $1\frac{3}{4}$ lbs. coal per hour; and the full value of this is about 10 horse-power, $[1\frac{3}{4} \times 5{\cdot}65 = 9{\cdot}9]$. Thus the I.H.P. developed in the cylinders of a good modern engine represents only $\frac{1}{10}$th the full value of the fuel; in other words, 10 per cent only of the energy due to the fuel is utilised in doing work, the other 90 per cent being lost.

B. Towing experiments for determining the ship's true resistance at various speeds, such as those conducted for the Admiralty by the late Mr. Froude and continued by Mr. R. E. Froude, show that the *effective* horse-power, by which is meant the

horse-power actually represented by the propulsion of the vessel at a given speed, is about half the total or *indicated* horse-power developed in the cylinders at the time, the other half being absorbed in various ways. This means that the effective horse-power represents only about 5 per cent of the full value of the fuel burnt.

Efficiency.—The efficiency of any machine, or of any process, is the proportion which the useful result bears to the expenditure; thus a machine which utilises half the energy expended upon it has an efficiency of one-half.

The whole propelling apparatus may be divided into four parts: the boiler which produces the steam, the steam-engine which develops power, the mechanism of the engine which transmits power to the screw, and the screw which applies the power to propel the ship.

1. *Boiler.*—The full heat-value of a pound of good coal (14,500 B.T.U.) would be sufficient for the evaporation of about 13 lbs. of water at present temperatures, each lb. requiring 1110 to 1120 units, from the temperature of the feed. The actual result in service boilers when steaming easily with Welsh coal is probably $8\frac{1}{2}$ to 9 lbs. water, which amounts to 65 to 70 per cent of the full value, the lower figure being much the more usual.

The efficiency of the boiler may therefore be taken as $\frac{6{\cdot}5}{10}$; in exceptional cases it may be $\frac{7}{10}$.

2. *Steam, or Steam-Engine.*—Since the steam brings from the boiler about 65 per cent of the heat due to the fuel, and the I.H.P. developed represents only 10 per cent, the efficiency of the steam is $\frac{10}{65}$ or $\frac{1}{6{\cdot}5}$. This figure may be directly obtained from the fact (ascertainable by measurement of the feed-water) that about 15 lbs. of steam are probably used per hour for each I.H.P. developed in the best service engines working at economical powers; and this represents a heat-expenditure per hour sufficient for $6\frac{1}{2}$ I.H.P., if all could be utilised, so that the efficiency is $\frac{1}{6{\cdot}5}$.

The fractions $\frac{6{\cdot}5}{10}$ and $\frac{1}{6{\cdot}5}$, multiplied together, make the $\frac{1}{10}$ which, as we have previously seen, represents the proportion $\frac{\text{Value of I.H.P.}}{\text{Value of fuel}}$.

Mechanism and Screw.—The combined efficiency of about one-half, which represents the value of the proportion $\frac{\text{Effective Horse-Power}}{\text{Indicated Horse-Power}}$, is the product of two efficiencies, viz. of the mechanism and of the screw, which are probably about $\frac{8}{10}$ and $\frac{5}{8}$ respectively.

The total loss of 95 per cent may be thus distributed:—

Calling full value of fuel 100.

		Loss in
Value of heat spent on steam is 65	Boiler	(100 − 65) = 35
	Steam	(65 − 10) = 55
„ I.H.P. developed is 10	Mechanism and Screw	(10 − 5) = 5
„ Effective horse-power is 5		Total loss 95

The principal sources of loss are as follows:—

Boiler.—Incompleteness of combustion; radiation; high temperature of funnel gases necessary to produce natural draught, and largely unavoidable with forced draught.

Steam.—Radiation; incompleteness of expansion; action of cylinder metal; heat in exhaust steam.

Mechanism.—Friction of working parts; load on air-pump.

Screw.—Frictional and edgeways resistance of water to movement of blades; augment of resistance; slip, and rotation of water.

Two of the above losses are entirely unavoidable, being necessary to the action of the machinery, viz. the slip of the propeller and the necessary heat in the exhaust steam. The others (excepting the first) can only be reduced, not altogether avoided, by any perfection in the arrangements. With draught produced by natural means, a large loss from the temperature of the funnel gases is also quite unavoidable; and, even where a fan draught is employed, some loss from this cause must of necessity be incurred, which is added to by the expenditure of steam necessary for driving the fans.

CHAPTER XIII

SCREW-PROPELLER AND PROPULSION

THE use of a screw as a propeller is obviously derived from its use in machinery as a means of producing motion in the direction of its axis. The *thread* on the screw engages with a corresponding thread on the inside of the nut, and rotation of one of these pieces relatively to the other must cause relative motion along the axis. The relative movement per revolution will be through a distance equal to the *pitch* of the screw, that is, the distance between consecutive turns of the screw-thread measured parallel to the axis.

Screws used in this way are frequently *double-threaded* where the forces exerted between the screw and the nut are considerable, and considerable travel per revolution (in other words, a coarse pitch) is desired. The screw has two threads, precisely alike, which run alternately, that is, at a distance of half a pitch apart. The pitch of the screw is the pitch of either screw-thread defined as above. Similarly three or four-threaded screws may be used.

The screw-propeller was tried experimentally, first with models, then with small boats, during the period 1835-38—the screw used being single threaded, and having a length, at first, of two or more complete pitches, subsequently reduced to one pitch.

In the latter year a larger boat, the *Archimedes*, was built specially to exhibit the powers of the screw, which she did with the greatest success during 1839-40. Her first screw was single threaded, and one pitch in length; this was afterwards replaced by a double-threaded screw of half a pitch.

The *Rattler* was then built by the Admiralty to test the screw on a larger scale. She was extensively tried during the years 1844-45, and with such satisfactory results as to practically

determine the substitution of the screw for the paddle-wheel in war-vessels. Her screw was at first, like that of the *Archimedes*, a double-threaded screw of half a pitch in length; but, as a result of repeated trials, the length was gradually reduced, until it became one-sixth only of the pitch, which proportion gave the best performance.

By the length of the screw is meant the extreme length in a fore-and-aft direction occupied by the blades when at the working angle. Supposing a double-threaded screw of a complete pitch to be divided into six parts of equal length, each piece will be similar to the piece E, shown in front and end view, Fig. 100, and will represent the elementary form of a two-bladed screw-propeller, the two blades carried on opposite sides of the axis being portions of the two screw threads. A three or four-threaded screw, similarly divided, would produce a three or four-bladed screw-propeller. The length taken in the latter cases would be only $\frac{1}{8}$ to $\frac{1}{10}$ pitch.

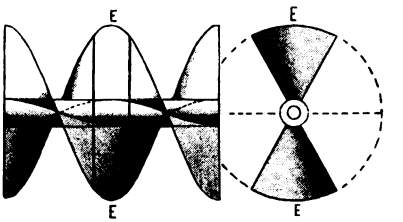


FIG. 100.

Three-bladed and four-bladed screws are preferred for general work at sea. They are more steady in their action in a sea-way, produce less vibration, and are less affected by the loss of a blade than are two-bladed screws, though the latter appear to have a slight superiority in propulsion in smooth water. Two-bladed screws were formerly widely fitted in the service, where it was desirable, in ships with sail power, to be able to lift the screw out of the water, or to feather the blades, in order to reduce the resistance when sailing, but these arrangements have become obsolete.

Diameter.—The diameter of the screw is the diameter of the circle struck out by the tips of the blades. This is, of course, limited by the draught of water; further, the upper blade, when upright, should be immersed by not less than $\frac{1}{10}$ diameter. Moreover, a large diameter, though good in other respects, involves increased loss by blade friction, and may thus, beyond a certain limit, lead to inefficiency.

Pitch.—The pitch is directly related to the required speed and the revolutions. For instance, for a given speed a fine pitch

involves high revolutions; this leads to economy of weight and space in the engines, but a limit in this direction is set by the increased loss by blade friction and the risk of damage to the machinery. With the increased speeds of ships, however, the revolutions are much higher than they were formerly, ranging from about 100 in battle-ships to 250 in the larger gunboats. The pitch is usually from 1·1 to 1·4 times the diameter.

The original form of the screw-propeller, produced as previously described, was soon found to be capable of improvement, notably by narrowing the outer part of the blade. The most marked improvements came with the introduction of Griffiths' screw during the period 1855-60; and, on the whole, the features of this screw have been found by experience to be those most conducive to success. These are (1) a large spherical boss, made much larger than would be necessary for the purpose of securing the propeller to the shaft, so as to remove the parts of the blades near the axis, which parts, lying too nearly in the fore-and-aft direction, were of little use in propulsion, and wasted power by churning the water; (2) the blades diminishing in width towards the tips; (3) the blades separable from the boss; (4) the pitch alterable.

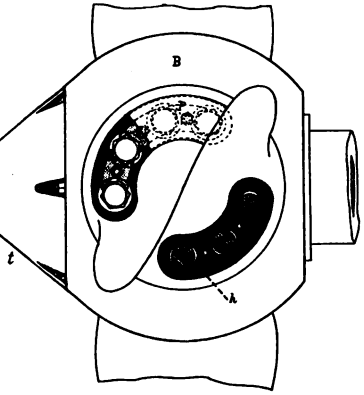
The screw in general use in the service, and very largely also outside, is a more or less modified Griffiths' screw. The details are shown in Figs. 101, 102, and 103.

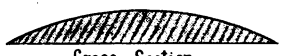
Material.—In the service, and in the best practice outside, the propeller is entirely of gun-metal, or some other description of bronze; this avoids the corrosion which occurs with both cast iron and steel. Compared with cast iron, also, bronze allows the use of thinner blades, which revolve with less resistance.

Blades.—The blades are generally of elliptical outline if we assume them to be untwisted, or *expanded* (dotted outline, Fig. 101), that is, speaking approximately, if the actual breadths of the blade at various distances from the centre be laid down on a flat surface in their proper positions. The broadest part of the expanded blade is usually at about half the extreme radius of the screw, though considerable variation exists both in outline and proportion, and appears to be consistent with efficiency.

In the service the blades are always made separate from the boss, and are secured to it by means of a flange and bolts, fitted as will be presently described. The blade is gradually reduced

FIG. 103.





Cross - Section.

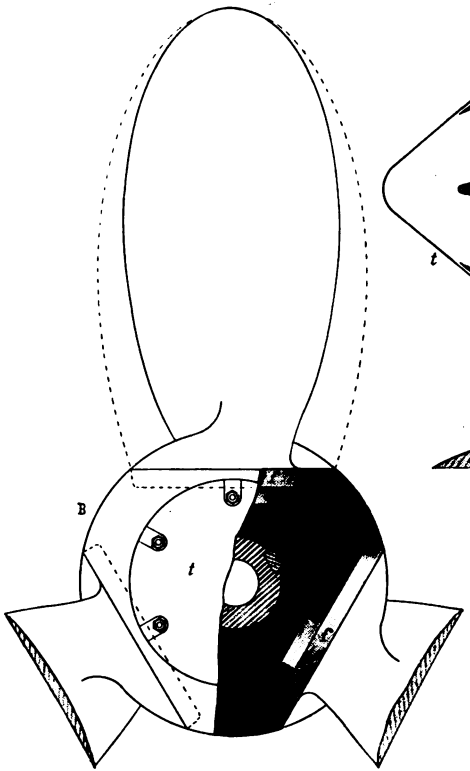


FIG. 101.

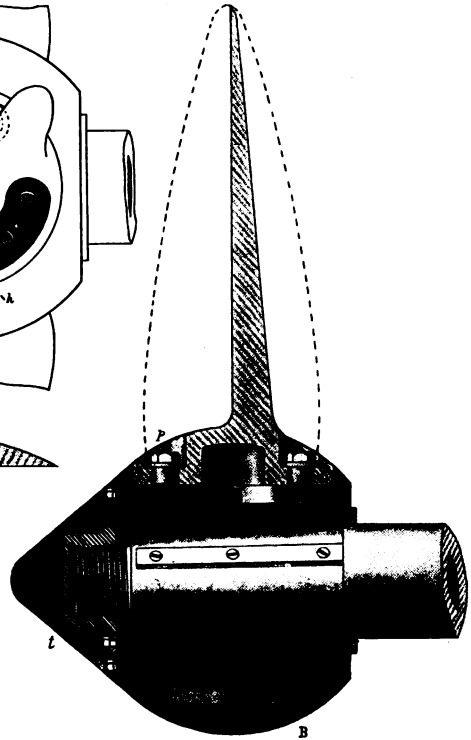


FIG. 102.

in thickness towards the tip, as in Fig. 102; and the cross section is of the form shown separately, being designed to secure the necessary strength with fairness of form for cleaving the water. The *forward* face or *back* of the blade is therefore not a true screw surface.

Boss.—The boss, B, of about $\frac{1}{4}$ the entire diameter of the screw, is fitted tightly on a tapered part at the after extremity of the shaft, to which it is secured for rotation by a key. The boss is kept in place by a nut screwed up behind it, and the nut is secured by a cotter or by a keep-plate. The boss is of circular section transversely, and, when complete, is pear-shaped longitudinally; for a conical *tailpiece*, *t*, is fitted to the after part, enclosing the tail end of the shaft, and preventing the nut from being fouled by ropes. The addition of the tailpiece reduces the resistance to the passage of the boss through the water.

To reduce the resistance offered to the revolution of the screw, the flanges, *f*, for securing the blades, are let into recesses in the boss, as in Figs. 101, 102, and their outer surfaces preserve the form of the boss; the heads of the bolts also, by which the blades are secured, are let into recesses in the flanges, as in Fig. 102, these recesses being afterwards covered by plates, *p*, curved to the form of the boss. Keep-plates prevent the bolts from slacking back.

Variation of Pitch.—To allow the pitch to be varied the bolt-holes, *h*, in the flange are elongated, as shown in Fig. 103, so that the blade may be turned in its seat and fixed in different positions, suitable filling pieces being fitted to the spaces on either side of the bolt in each position. The object of varying the pitch is: (1) to enable the best pitch to be found by trial; (2) to enable the pitch to be reduced when the boilers are worn, and the working pressure has consequently to be reduced. The steam of lower pressure occupies a greater volume per pound; and to enable the same quantity, or *weight*, of steam to be used, and so keep up the power and speed as far as possible, the revolutions require to be increased to provide increased cylinder capacity. The higher revolutions can only be produced by using a finer pitch.

Action of Screw-Propeller.—When the screw-propeller is revolved, say for ahead, the blades press upon the water in contact with them as they would upon the thread of a nut. The

result is that the screw moves forward, while the water, which only imperfectly resists the pressure, is driven backward, and is also rotated and driven outward from the axis of rotation. Only the directly sternward movement of the water and the forward movement of the ship are the result of useful effort: any other movement in the water represents waste work.

If the water acted on did not yield to the pressure, the screw would advance through it by a distance equal to the pitch per revolution; the advance, in feet per minute, would therefore be found by multiplying pitch by revolutions per minute; this product is called the *speed of the screw.*

The water acted on is, however, driven back relatively to the body of water immediately around it, with which it was previously in common; the advance of the screw (and ship), *relatively to this body of water*, is therefore always less than the speed of the screw. This backward movement of the water acted on, relatively to the immediately surrounding water, constitutes the *real slip* of the screw, though it is not what is generally known as slip.

As usually defined, slip is the difference between the *speed of screw* defined above and the *speed of ship relatively to still water.* It is expressed either in feet per minute, or, more frequently, as a *percentage of the speed of screw.*

Thus:

$$\text{Slip} = \text{Speed screw} - \text{Speed ship, in feet per minute,}$$

$$\text{and, Slip per cent} = \frac{\text{Speed screw} - \text{Speed ship}}{\text{Speed screw}} \times 100.$$

Example.—Given pitch 17 feet, revolutions 100, speed of ship 15 knots, find slip per cent.

$$\text{Speed screw} = 17 \times 100 = 1700 \text{ ft. per min.}$$

$$\text{Speed ship} = \frac{15 \times 6080}{60} = 1520 \quad ,,$$

$$\text{Slip} = 1700 - 1520 = 180 \quad ,,$$

$$\text{and, Slip per cent} = \frac{180}{1700} \times 100 = 10{\cdot}6.$$

In practice, with an efficient propeller, the slip, found as above, is usually 10 to 15 per cent. It may sometimes, however, prove to be much less than usual, for the following reasons:—

The slip, as defined above, is only part of the total or real slip. It can only represent the whole of the slip on the assump-

tion that the screw acts upon still water, which is not practically the case. The water on which the screw acts is really part of a current moving in the same direction as the ship, relatively to still water, having been set in motion by the frictional drag exerted upon it by the immersed skin of the ship.

The backward thrust of the screw has the effect, first, of arresting this forward motion in the water acted on; and, second, of impressing upon it more or less actual backward motion (forward and backward here meaning relatively to *still water*); and it is only the second part which appears as the result of the calculation for slip.

Now under certain conditions, such as foulness of the ship's bottom, or unsuitability of the form of ship for the speed at which she is being driven, the velocity of the following current in which the screw works may become greater than usual; so that the first part of the total slip becomes unusually great, and the second part, commonly known as the slip (better distinguished as the *apparent* slip), correspondingly small; the total or *real slip*, which is the sum of the two, being alone a definite quantity dependent on the screw's action.

Whenever the slip is found to be abnormally small the conditions of propulsion are defective, as will be seen on comparing the speed with the power used to obtain it, when it will be found that the power is excessive. The reason for this is that the forward current is produced at the expense of the ship's engine power, and the expenditure is in these cases greater than usual; and though the energy so spent is partially recovered by reason of the screw working in the current instead of in still water, it is only partially recovered, because some of the water set in motion escapes the screw altogether.

Twin-Screws.—Twin-screws have been applied to all naval vessels built during the last ten or twelve years, with the exception of some gunboats, and to all battle-ships for more than twenty years past; and their use has of late years become extended to large passenger and trading vessels of the mercantile marine. Some of their advantages are obvious; the most prominent being that the ship is less liable to suffer total disablement of her propelling machinery, and that she possesses superior handiness in manœuvring, and a means of steering apart from the ordinary steering gear. These are important advantages in any

ship, and they are especially valuable in a war-vessel. Further, twin-screws favour the use of the vertical engine in war-vessels, by reducing the height necessary for the engines, and so facilitating their protection. The two sets run at higher revolutions than would a single set developing the same total power; and with higher revolutions a shorter stroke may be used, and the height of the engines, which is approximately a multiple of the stroke, is consequently reduced.

Apart from the practical advantage of using the vertical type of engine, the vertical twin-screw arrangement has the great advantage of leaving the middle line clear for a fore-and-aft bulkhead dividing the engine-room space into two water-tight compartments. This desirable end could not be attained with a single screw.

The propulsive efficiency is believed to be at least as great in twin-screw as in single-screw ships.

Screw and Paddle-Wheel.—Besides the very obvious advantages of the screw in comparison with the paddle, especially in war-ships, there is a marked advantage which requires reference. The pitch of a screw is always very much less than the circumference of a paddle-wheel large enough to be efficient for the same vessel; the result is that screw engines revolve at much higher revolutions than paddle engines, and this leads to a considerable reduction in weight and size of engines for a given power, as was stated in the introductory chapter. As a propelling instrument the screw is more efficient than the paddle-wheel, especially in long ships, and on long voyages where considerable changes of draught occur. Its efficiency arises from the fact that it acts on a very large body of water, probably the whole of the water which comes within the circumference of the screw's disc; and it is little affected by changes of draught. It is also well placed, especially the single screw, for acting on the following wake.

On the other hand, there is a disadvantageous effect arising from the action of the screw, known as "Augment of Resistance," which consists in a virtual increase of the ship's resistance, due to the water-pressure on the stern of the vessel being diminished by the backward action of the screw on the water, the water-pressure forward being thus left partially unbalanced. This detracts from the efficiency of the screw's action; and it has accordingly been noticed, in the statement on efficiency, as an

element of inefficiency in the screw. It is less with twin-screws than with a single screw. This advantage, however, in twin-screws is partly balanced by the resistance due to the brackets which support the screws.

River steamers, and other vessels of light draught working in shallow waters, especially if they require a high speed, are still commonly propelled by the paddle-wheel. It is difficult to deliver the necessary power by means of screws so small as would be available in such vessels, though it may be done with fast-running machinery, as in torpedo-boats.

Resistance to Ship's Progress

In smooth water, and at moderate speeds, the resistance of a well-formed ship is principally that due to friction between the water and the immersed skin of the ship; and its amount depends on the area and length of the immersed surface, and on its condition as to roughness, as well as on the speed. Assuming the conditions other than speed to remain the same, frictional resistance will vary as the square of the speed; and the experiments on resistance before referred to show that at moderate speeds this law approximately expresses the connection between resistance and speed in the case of a ship.

Beyond a certain speed, however, in any given ship, the surface water becomes more seriously disturbed than at lower speeds by the passage of the ship; waves are raised to a marked and increasing degree, and the *wave-making* resistance thus added on rapidly increases, and causes the total resistance also to increase in a rapidly increasing ratio as the speed advances. The actual speed at which this change commences in any given ship depends on the form and proportions of the hull, and especially on the lengths of entrance and of run, being a higher speed with greater length and finer form of ship. Beyond this point the speed can no longer be considered moderate for the particular ship.

Relations between Power, Speed, and Consumption

Power and Speed.—If total resistance = R lbs., and speed of ship = V feet per minute, then evidently R × V = effective work of propulsion per minute in ft.-lbs.; and

$$\frac{R \times V}{33{,}000} = \textit{Effective Horse-Power.}$$

If we assume that the effective horse-power is a definite proportion of the indicated horse-power at all speeds, which is only approximately true, the above expression shows that I.H.P. is proportional to $R \times V$. Supposing, further, that we are dealing with moderate speeds, so that R may be assumed to vary as V^2 (or $R \times V$ as V^3), we have

$$\textit{I.H.P.} \propto V^3 \quad . \quad . \quad . \quad (a).$$

This relation between power required and speed produced holds approximately so long as we keep to speeds which are moderate for the particular ship; at the higher speeds the increase of I.H.P. with increased speed is more rapid, taking place according to an increasing power of the speed. Speaking generally, the law may be applied up to within say 2, 3 or 4, and 4 or 5 knots of full speed in 13, 16, and 20 knot ships—the actual figure depending on the form and proportions of the ship in relation to the speed, as before mentioned. From this point the power increases more rapidly, and the last knot of full speed is obtained by an increase of power in proportion to the fourth, fifth, or even sixth power of the speed, according to the suitability of the vessel for the speed at which she is driven. Again, in going to very low speeds the decrease of I.H.P. with decrease of speed becomes, after a certain point, less rapid than is given by the law; the reason being that a part of the engine friction remains practically constant, so that its effect becomes disproportionately great at the lowest powers.

Consumption per Hour and Speed.—The relation (*a*) may be modified in form by assuming that the coal consumption per hour is proportional to the I.H.P. developed; in other words, that the consumption per hour for each I.H.P. developed is the same at all powers. This, again, is only approximately true; for, as has been previously stated, the consumption per hour per I.H.P. increases both at very high and at very low powers; nevertheless it is fairly constant for a large range of intermediate powers.

This assumption gives us the relation—

$$\textit{Consumption per hour} \propto V^3 \quad . \quad . \quad . \quad (b).$$

The relation (*b*) is of course, like (*a*), inapplicable to the

highest speeds, and would also be in error if applied to very low speeds. In either case, too, the error in the relation (*b*) would be greater than in (*a*), as will be seen from the facts stated. In decreasing a very low speed, and especially in increasing a high speed, the coal estimate given by the relation (*b*) would therefore be too small.

Examples on (*a*) and (*b*):—

1. Given that 1500 I.H.P. produces speed of 10 knots, find I.H.P. necessary for speed of 12 knots, both being moderate speeds for the ship.
2. A ship burns 30 tons per day at 10-knot speed; find probable consumption if speed be increased to 12 knots, both being moderate speeds for the ship.

The working is obvious:

$$(1) \quad \frac{x}{1500} = \frac{12^3}{10^3}, \text{ or } x = 2592 \text{ I.H.P.}$$

$$(2) \quad \frac{x}{30} = \frac{12^3}{10^3}, \text{ or } x = 52 \text{ tons per day.}$$

Consumption per Knot and Speed.—Returning to the investigation, two useful relations may be deduced from (*b*) without any further assumptions. If we divide each side of (*b*) by the speed, we have

$$\frac{\text{Consumption per hour}}{\text{Knots per hour}} \propto V^2.$$

That is, *Consumption per knot* $\propto V^2$ . . . . (*c*).

Consumption, Speed, and Distance.—Again, if we multiply each side of (*c*) by any given distance run at speed V, we have,

Consumption over any given distance at speed V
$\propto V^2 \times$ *Distance* . . . . . (*d*).

The relations (*c*) and (*d*) enable a direct comparison to be made between coal consumption and useful result obtained (*i.e.* distance made good) at any given speed, and so bring directly into view the practical efficiency of the entire performance.

By means of relation (*d*), having information as to previous voyages, an estimate may be formed of the probable coal expenditure on any proposed voyage at any proposed speed (allowance being afterwards made for possible adverse circumstances), and the best speed for the object in view chosen.

The laws (*c*) and (*d*) are of course, like (*b*), not applicable to the extreme speeds.

Examples on (*c*):—

1. A high-speed ship burns 2 tons per hour at 12 knots, at what speed should she steam to make a distance of 1200 knots, expending 240 tons on the engines alone?

Let x = required speed.
Consumption is to be $\frac{240}{1200} = \frac{1}{5}$ ton per knot,
and consumption at 12 knots is $\frac{2}{12} = \frac{1}{6}$ ton per knot.
Therefore, since consumption per knot varies as speed^2,

$$\frac{x^2}{12^2} = \frac{\frac{1}{5}}{\frac{1}{6}}, \text{ or } x^2 = 172{\cdot}8$$

and $x = 13{\cdot}1$ knots.

2. A vessel is distant 400 knots from port and has 160 tons of coal available. The consumption at 8 knots is 1·6 tons per hour. What is the shortest time in which the vessel can reach the port under steam?

At 8 knots the consumption is $\frac{1{\cdot}6}{8} = {\cdot}2$ ton per knot.
The ship has available 160 tons for 400 knots, or $\frac{160}{400} = {\cdot}4$ ton per knot.
If x = speed which must be adopted,

$$\text{then } \frac{x^2}{8^2} = \frac{{\cdot}4}{{\cdot}2} = 2,$$

$$\therefore\ x^2 = 128, \text{ or } x = 11{\cdot}3 \text{ knots},$$

$$\text{and time required} = \frac{400}{11{\cdot}3} = 35\tfrac{1}{2} \text{ hours}.$$

Example on (*d*):—

If a ship burn 250 tons in steaming 1000 knots at 12-knot speed, how much will she require for a distance of 1500 knots at 10 knots, assuming equally favourable conditions, and excluding auxiliary purposes in both cases?

If x = coal required,

$$\text{Then, by relation } (d),\ \frac{x}{250} = \frac{1500 \times 10^2}{1000 \times 12^2} = \frac{150}{144}$$

and $x = 260$ tons.

Curve of I.H.P. and Speed.—It has been stated that the above laws are approximate only, and of limited application. It is obvious also that many circumstances may act to modify the result—as, for instance, the weather, the condition of the ship's

bottom, the quality of the coal, and the condition and management of the engines. So far as these conditions are known, allowance must be made for them according to judgment and experience, unforeseen circumstances being provided for by a prudent margin. It may well appear, indeed, from all that has been said, that the laws are very defective; still, in the absence of definite information as to the performance of the ship at the

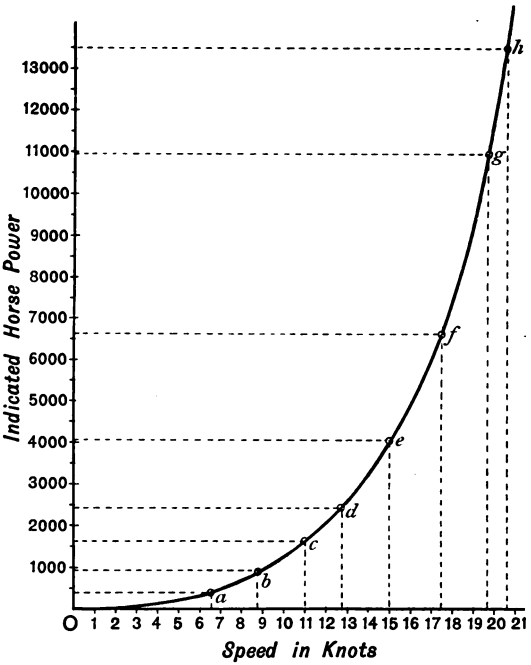


FIG. 104.

speed proposed, they supply a useful basis of calculation, which requires only to be used with discretion.

Definite information, however, may be obtained from speed trials of the ship, such as are now carried out in naval vessels, either when new, or as opportunity offers subsequently, the results being plotted so as to obtain a curve of I.H.P. and speed.

Such a curve is shown in Fig. 104. The base line is divided to represent speeds, the vertical line to represent power. The result of each trial is shown by the position of a point, such as a, b,

$c \ldots h$, the vertical height from the base line, and the horizontal distance from the zero of speed, corresponding respectively to the power and the speed obtained on the trial. Having a sufficient number of such points, a free curve is drawn passing through them, and through the zero point, O. This curve once obtained, embracing the highest and the lowest speeds practicable, the I.H.P. corresponding to any proposed speed may be at once found by simple measurement; and the approximate consumption per hour at various powers being also known from the engine-room register, problems such as those given in the examples may be solved with confidence and accuracy for all speeds, from the highest to the lowest.

Auxiliary Purposes.—In addition to the coal used for the main engines, there is the coal used daily for auxiliary purposes, such as steering, electric lighting, distilling, etc. Of this expenditure it is highly desirable that a careful estimate be made, so that a proper allowance may be made for it. In some ships it may amount to an addition of, say, 8 to 10 per cent of the coal expenditure for main engines at half power.

Most Economical Speed.—By this term is meant the speed at which, apart from other considerations, the ship can steam a given distance with the least expenditure of fuel. It is, in fact, the speed at which the coal expended per knot is the least; or, in other words, the distance run per ton of coal is the greatest.

Referring to law (*c*), it appears that as we reduce speed we reduce the expenditure per knot in a more rapid ratio; and it is obvious that if this law continue to hold at all speeds, the best result will be obtained at the slowest possible speed at which the ship can be driven.

It is well known, however, that this is not the case. The expenditure per knot diminishes with the reduction of speed only until a certain speed is reached, after which it begins to rise as the speed is further reduced. The speed at which this change occurs can only be ascertained in any given ship by actual trial of the ship at various speeds, as opportunities offer for making the trials under favourable circumstances. Each trial ought to be, if possible, of 24 hours' duration at one continuous speed, to allow a fair estimate of the coal consumption to be made, and with a fair

condition both of sea and of ship's bottom. These conditions should be as similar as possible on the several trials.

The value of the fraction $\frac{\text{Coal expenditure per hour}}{\text{Speed}}$ being found in each case, the speed at which this fraction is a minimum will be the most economical. In a high-speed ship the economical speed corresponds usually to $\frac{1}{10}$ to $\frac{1}{15}$ full power; it is frequently about 7 knots.

The expenditure above considered is that for the main engines only; and the result is the speed most economical so far as that expenditure is concerned. But where, as in war-vessels, there is a large daily expenditure of fuel for auxiliary purposes, it is obvious that this should be taken into account in finding the speed at which the ship will steam the greatest distance with the coal on board. The speed most economical for the engine expenditure alone will be too low; the real economical speed, everything included, will in fact be that which gives the least value to the fraction $\frac{\text{Total expenditure for all purposes in given time}}{\text{Speed}}$. This will give a higher speed than that found from the previous fraction, as will appear from the consideration that the constant addition of a daily expenditure at all speeds will have a greater effect in increasing the proportion of expenditure to speed at the lower speeds than at the higher.

A graphic method may be usefully employed in examining and illustrating this matter.[1] In Fig. 105 the curve *ab* is drawn through points representing the results of trials at various speeds, the speed being represented by horizontal measurements from OY, and the coal expenditure by vertical measurements from OX.

If a tangent to the curve, drawn from O, touch it in E, then EC or OS is the most economical speed, and ES the corresponding consumption.

It is evident that the ratio $\frac{\text{Expenditure}}{\text{Speed}}$ is less at the point E than at any other point on the curve, because E is also a point on the straight line, and the ratio $\frac{ES}{OS}$ is constant for all points

[1] See paper on "Coal Consumption in Ships-of-War," read by Staff-Engineer W. H. Riley, R.N., before the Royal United Service Institution, June 1893. *Proceedings R.U.S.I.*, vol. xxxvii.

on the straight line, and is obviously less for them than for points in the curve above.

It will be seen, on examination of such a curve, that the point E may be shifted to some extent about its present position without materially affecting the above ratio, that is, without any material loss of economy; so that although E gives the most economical speed, there is a certain range of speed for which the

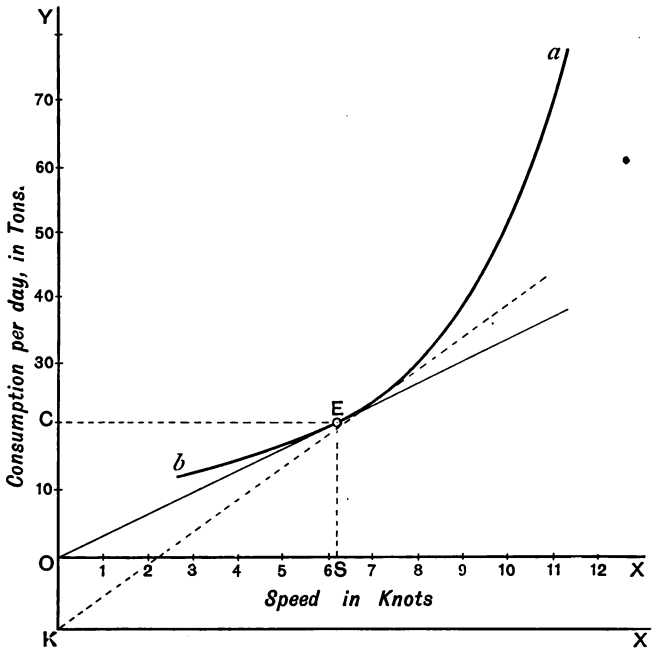


FIG. 105.

falling off in economy would be very slight. Further, if the line OE be drawn to cut, instead of touching, the curve, it will be seen that there are two speeds, above and below the economical speed respectively, giving the same degree of economy; of these the higher speed would, of course, be generally preferable.

In order to include any given constant daily expenditure for auxiliary purposes, it is only necessary to set off a distance OK below O to represent this expenditure, and then to draw the

tangent to the curve from the point K instead of from O. The point so found will be higher on the curve than the point E, and will represent the minimum value of the ratio $\frac{\text{Total Expenditure}}{\text{Speed}}$.

Again, the effect of a current of known strength may be allowed for by setting off a corresponding distance along OX or KX, as the case may be,—to the right if adverse, to the left if favourable,—and drawing the tangent from the point so found.

THE END

Printed by R. & R. CLARK, *Edinburgh*